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TATA**

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**National Aeronautics and
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HIGH INTENSITY ACOUSTIC TESTS OF A THERMALLY STRESSED ALUMINUM PLATE IN TAFA

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ABSTRACT

An investigation was conducted in the Thermal Acoustic Fatigue Apparatus at the Langley Research Center to study the acoustically excited random motion of an aluminum plate which is buckled due to thermal stresses. The thermal buckling displacements were measured and compared with theory. The general trends of the changes in resonance frequencies and random responses of the plate agree with previous theoretical prediction and experimental results for a mechanically buckled plate.

SUMMARY

An investigation was conducted in the Thermal Acoustic Fatigue Apparatus to study the random motion of a buckled aluminum plate exposed to heat and intense acoustic loads. Plate buckling was due to thermal stress. The plate was exposed to noise levels up to 160 dB and temperatures to 250°F. Two different thermal boundary conditions of the plate were studied; one condition with the plate clamped in a steel frame and the other with the plate insulated from the steel frame. For the second condition, the temperature distribution and buckling deflection were considerably different than for the first condition. The acoustic response was also significantly different for the two boundary conditions. The general trends of the changes in resonant frequencies and random response of the plate agree with previous theoretical prediction and experimental results.

INTRODUCTION

One of the problems to be encountered by future hypersonic aircraft such as the National Aerospace Plane (NASP) will be the high thermal and acoustic environments to which the aircraft will be subjected during a significant portion of the

flight envelope. The heating will cause internal stress within the structure and possibly buckling of external plates. Acoustic fatigue of plates at elevated temperatures were investigated on metallic structures by Schneider¹ and on composite structures by Jacobson², but the non-linear characteristics of acoustic response associated with a buckled state were not studied. High intensity acoustic loads can cause snap-through (oil canning) of buckled plates.³ Some research on acoustic response has been conducted on snap-through of aluminum plates by uniaxial compression (mechanical)^{4,5} with no research on heating effects. The Thermal Acoustic Fatigue Apparatus (TAFA) at LaRC has a limited capability to study this problem of combined heating and acoustic loads.

This paper will present the results of an investigation using a thermally stressed aluminum plate. For one boundary condition, the plate was restrained on all four edges with steel mounting brackets. For the second condition, the plate was insulated from the steel brackets. Results include buckling due to thermal loads, responses due to acoustic and thermal loads, and snap-through. A comparison of the results with theory is made. In addition, a description of TAFA is given.

EXPERIMENTAL METHOD

Thermal Acoustic Fatigue Apparatus

The Thermal Acoustic Fatigue Apparatus (TAFA) is a grazing incidence, high intensity noise facility with capability of sound pressure levels from 120 to 160 dB both sinusoidal and random in the frequency range of 40 to 500 Hz (Figure 1). The noise source is two 30,000 watt acoustic modulators using filtered pressurized air. The sound is coupled to the test section by an exponential horn with a 27 Hz low frequency cut-off. Test panels or plates with maximum dimensions of 12" x 15" can be mounted in one wall of the test section. Heat is provided by a bank of 12-2500 watt quartz lamps (Figure 2) located in the wall opposite and 12" distant from the test

specimen. Heat is controlled to either a percentage of the heat output available, or a fixed temperature at some point on the test specimen.

The thermal capability of the facility was determined with a 12" x 15" x 0.030" steel plate instrumented with high temperature thermocouples with stainless steel coverings over the lead-in wires (Figure 3). The photograph shows the plate after it had been subjected to the maximum temperatures. The darkened area surrounding the plate is the slightly charred insulation protecting the wall around the test specimen. The temperature on the plate was found to be a function of both the sound pressure level (SPL, dB) and the air pressure PSIG (gage pressure) as well as the heat source (Figure 4). At zero PSIG (no air flow) slightly over 1050°F was obtained at maximum power (30 kilowatts). At normal operating pressure (30 PSIG) and nearly maximum acoustic level (160 dB) 780°F was obtained. Note that the temperature decreased as the noise level increased, especially at the higher noise levels. This decrease is due partly to the increased air flow and partly due to increased convection resulting from high acoustic particle velocity. A temperature survey over the upper half of a steel plate is shown in Figure 5. Note that the highest temperatures were along the horizontal center line (6 inches down from top of plate). There is some temperature drop as the top of the plate is approached, and a greater drop as the leading or trailing edge of the plate is approached.

Experimental Set-Up

One aluminum plate with two mounting conditions (boundary conditions) was used in this investigation. Its plan form was 12" x 15" and was 0.063" thick (Figure 6). For one boundary condition, the plate was mounted in steel brackets clamping the plate on all four edges. Great care was required in tightening the brackets to assure that no buckling (or bi-axial stress) occurred. For the second boundary condition, the same plate was mounted with ten layers of 0.006" fiberglass tape around each mounting surface and then held in the steel brackets.

Instrumentation

The instrumentation on the aluminum plate consisted of both strain gages and thermocouples as shown in the photograph in Figure 6 showing front or sound incident side. Eight thermocouples were mounted on the front of the plate and three sets of strain gages were located on the front and back of the plate as shown in Figure 7. The outputs of the thermocouples were recorded on a multi-channel data-logging system. The strain gages were on each side of the plate to form one-half of a bridge completion unit. The gages in the center and one set of gages near the edge of the plate were wired to read bending and the other set of gages near the edge of the plate was wired to read tension. The outputs of the bridge completion units were both recorded on a multichannel analog data recorder and were also directed to oscilloscopes and to a real time data analyzer.

Microphones were used for determining noise level and spectra over the plate (Figure 6). These microphones were flush mounted in tubes (I.D. = 0.201") about 17 inches outside the wall of TAFA to move them away from the heat. Attached to the tubes just beyond the microphone locations were 50 foot coils of one-quarter inch copper tubing as shown in Figure 8. (NOTE: The photograph shows the locations for the tubes and microphones for a 12" x 12" test plate.) These coils were not plugged and reduced back reflections of noise from reaching the microphones.

Measurements and Analyses

Measurements were made of noise level, temperature, static deflection, and strain. The noise levels were determined as the arithmetic average between the two microphones. These indicated sound pressure levels (SPL) from each microphone were within 1 dB of each other and had very similar spectra. The spectra over a 500 Hz bandwidth obtained for the various noise levels are shown in Figure 9. Note that the spectra indicate lower levels between 200 and 250 Hz although the electrical input to the modulators is white noise in the range of 20 to 500 Hz (Figure 10).

The temperature distribution for the maximum temperature condition, 250°F at the center of the plate, was measured for only the top left-hand quadrant of the plate. The internal calibration of the thermocouple data-logging system gave consistent results on a daily basis. Unfortunately, the bonding of a thermocouple occasionally failed rendering the thermocouple useless.

The static strain levels under elevated temperatures could not be determined from the strain gages because the output voltage from the strain gages as expected was a function of both temperature and buckling configuration. In addition, the combined steady state and dynamic outputs of the bridge completion units were occasionally over one volt root mean square (rms). Thus they were adjusted (rebalanced) such that the dynamic outputs were less than one volt rms and could then be recorded on the tape recorder. The static deflection was measured with a dial indicator on the side of the plate opposite the heat.

All appropriate calibrations were made to insure that the sound pressure levels, temperature and strain measurements were accurately measured. In addition to using a sound level calibrator on the "tube" microphones, sound pressure levels were measured with a third calibrated microphone in TAFA for the random noise that would be used in this investigation. Comparison of the results of these noise measurements of the tube microphones and the standard microphones showed excellent agreement. In addition, the tube microphones were installed and calibrated one at a time in a one inch pipe opposite a calibrated microphone. At one end of the pipe was an acoustic driver and the other end was open. Again, excellent agreement between the calibrated microphone and the tube microphone was obtained.

The strain gages were calibrated with a one megohm resistor. The calibration factor for the output of each of the strain gage completion units was 0.4706 micro-strain per millivolt ($\mu\epsilon/\text{mV}$). A number of functions, namely probability density, power

spectra, and rms strain response were obtained from a fast fourier transform analyzer, both in real time and from tape recordings. Time histories were also recorded.

RESULTS

Temperature distributions over one quadrant of the aluminum plate with no insulation and with insulation are shown in figures 11 and 12. For the uninsulated plate, the following was determined: 1) the average temperature rise above the edge temperature (figure 13); 2) center deflection due to plate temperature (figure 14); 3) acoustic response as a function of plate temperature (figure 15); and 4) strain response spectra at various temperatures (figures 16-18). In addition, comparisons of the bending and tensile PSD's (figure 19) are made as well as showing the probability density distribution of bending strain responses (figure 20).

For the insulated plate, the following was determined: 1) the average temperature rise above the edge temperature (figure 21); 2) center deflection due to plate temperature (figure 22); 3) acoustic response as a function of plate temperature (figure 23); and 4) strain response spectra at various temperatures (figures 24-27).

DISCUSSIONS

The discussion will be given in two sections - one section for the uninsulated plate and the other for the insulated plate. Thermal effects, buckling displacement, bending and tensile strain measured at the edge of the plate, and probability density will be discussed for both boundary conditions. This section will be concluded with a comparison of the two conditions.

Uninsulated Plate

Thermal effects.- The temperature distribution for one quarter of the uninsulated plate for a center temperature at 250°F is shown in Figure 11. The temperature difference between the center of the plate and the edge of the plate was about 110°F. If a parabolic temperature distribution is assumed, the average rise of

plate temperature above the surrounding edges was approximately 56°F. This temperature is used to estimate the thermal buckling instead of using the temperature above ambient (87°F) because the surrounding edges are no longer at ambient temperature. The average rise of temperature is plotted against center temperature in Figure 13.

Thermal buckling displacement.- From references 1 and 6, the temperature rise for the start of buckling is 9.6°F for the boundary conditions of this study. This rise in temperature corresponds to a center temperature of 114°F (Figure 13). The buckling mode shape is the 1,1 mode. The theoretical and experimental results for center deflection due to buckling is shown in Figure 14. The agreement is very good for higher temperatures (up to 250°F) but poor near the buckling temperature. This disagreement is attributed to the presence of imperfection (non-flatness) in the plate. The corresponding surface strain due to bending in the buckling mode is calculated from the experimental deflection by assuming the 1,1 mode since direct strain measurement cannot be made as mentioned earlier. The scale for the calculated strain is shown on the right hand side of Figure 14.

Dynamic bending strain.- Figure 15 shows the variation with temperature of rms dynamic bending strain resulting from broadband acoustic excitation. At 160 dB SPL, the maximum response of 571 microstrain is obtained at 120°F. This trend of variation of rms strain agrees with previous theoretical and experimental results.^{4,5} Inspection of the time history of the response at 120°F, (Figure 15, insert A) shows that at 160 dB SPL, there is persistent snap-through motion. Only intermittent snap-through motion is found at 150°F, (Figure 15, insert B), and no snap-through motion is found at 200° and 250°F (Figure 15, insert C). The buckling displacement of 150°F (figure 14) is about twice the thickness of the plate. For higher buckling deflection at 200°F and higher, the overall rms strain responses decrease considerably due to the stiffening effect of the curvature.

Bending strain spectral response - ambient (87°F).- The fundamental plate frequency at 130 dB SPL was found to be 103 Hz (Figure 16) which is lower than the theoretically predicted value of 120 Hz. This may be due to the non-ideal clamping condition. The strain levels for the second and third modal resonances, at 190 and 302 Hz respectively, are several orders of magnitude lower than the fundamental resonance. At 140 dB SPL, the resonant frequency decreases slightly and there is a significant peak at 1 Hz compared to the fundamental resonant peak response. This may be due to the curvature effects of the considerable amount of imperfection in the plate. At 160 dB SPL, the resonant frequency peak broadens with the center of the peak response at 120 Hz.

Bending strain spectral response - 120°F.- At this higher temperature the peak response for 140 dB SPL was at 200 Hz (Figure 17). The resonant frequency decreased to 170 Hz as SPL increased to 150 dB SPL. At 152 dB SPL, the peak in the modal response at 150 Hz is less than the peak at 150 dB SPL (170 Hz). In addition, there is a dominant peak in response at one Hz which is due to the intermittent snap-through. This intermittent motion results in low frequency and large amplitude vibration. The peak at 105 Hz dominates the response at 160 dB SPL and the response at one Hz decreases from that observed at 152 dB SPL because the snap-through motion is nearly continuous. For persistent snap-through motion the rms bending strain response was only 20 percent higher than that of the flat plate. This small increase resulted because the 160 dB SPL can only excite persistent snap-through motion of the thermally buckled plate at 120°F, for which the center deflection is only 62 percent of the thickness.

Bending strain spectral response - 150°F.- Modal peak responses are found at 140 Hz for 135 dB and 200 Hz for 145 dB (Figure 18). There is a very dominant peak at two Hz for 160 dB SPL. The response is greater than that found at one Hz at 120°F, for 162 dB, because of the larger snap-through motion.

Tensile strain spectral density.- The tensile strain spectrum is compared with the bending strain spectrum in Figure 19 at 120°F and 160 dB SPL. The peak response of tensile strain was at 210 Hz, twice that of bending strain at 105 Hz and was also much more broadened. In addition, there is a peak tensile strain at five Hz. These two peaks occur because the tensile strain is proportional to the square of the dynamic displacement.⁸ The overall rms tensile strain is about one eighth of that of the bending strain.

Probability density.- Figure 20 shows the probability density distribution of the bending strain response at 120°F. At 152 dB SPL, there is one dominant peak but at 154 dB SPL, as snap-through motion starts, there is also a second peak to the right of the first peak at almost the same magnitude. The two peaks correspond to the two static positions around which the oscillations are centered. The peaks are not symmetric because the vibration motions are non-linear. The probability density between the peaks increases as the snap-through motion between the two static positions become more frequent (158 dB) and the probability density plot looks like a broad plateau at 160 dB. The above results of double peaks and plateau in the probability density of random response are new characteristics of non-linear behavior that have not been reported before.

Insulated Plate

Thermal effects.- The temperatures at the edge of the insulated plate are closer to the temperature at the center of the plate (Figure 12) than for the plate with the uninsulated edges. The average rise in temperature above the surrounding edges is thus lower and is plotted against center temperature in Figure 21. The center temperature for buckling is 150°F.

Thermal buckling displacement.- The experimental results for the deflections are much higher than the theoretical results based on a 100 percent clamped plate (Figure 22). This discrepancy may be due to the non-ideal clamping of the edges containing the flexible insulating material.

Dynamic bending response.- For 160 dB SPL, the response increases with temperature (Figure 23) because persistent snap-through motion can be excited (Figure 23, insert A). The maximum response may be found at a temperature higher than 250°F. For 150 dB SPL, the maximum response is found at 150°F and the response decreases at 200°F because only intermittent snap-through motion can be excited at 200°F (Figure 23, insert B). For 200°F and 140 dB SPL, the oscillation is non-symmetric (Figure 23, insert C) with only occasional snap-through.

Bending strain spectral density - ambient (89°F).- The initial fundamental frequency at 130 dB SPL is 107 Hz (Figure 24). There is an increase in resonant frequency and a broadening of the peak frequency at 160 dB SPL.

Bending strain spectral density - 150 °F.- The fundamental frequency is initially 121 Hz at 130 dB SPL (Figure 25) and decreases to 98 Hz at 150 dB SPL, for which intermittent snap-through is found. The low frequency peak at one Hz is not very high compared with the peak at 98 Hz, showing that the magnitude of the snap-through is small. At 160 dB SPL, the peak response is found at 120 Hz again and the response peak at one Hz has disappeared.

Bending strain spectral density - 200°F.- The fundamental frequency is initially 132 Hz (Figure 26) and decreases to 105 Hz at 150 dB SPL, for which intermittent snap-through is indicated by the peak response at two Hz. This low frequency peak response disappears at 160 dB SPL.

Bending strain spectral density - 250°F.- The fundamental frequency is initially 139 Hz (Figure 27) and decreases to 118 Hz at 150 dB SPL. The spectra are similar to those of 200°F except that the peak response at two Hz is higher and the

modal response peak is broader. Therefore, the broadening of the modal response peak of persistent snap-through motion is greater when the buckling displacement is larger.

Tensile strain and probability density.- The observed trends on the uninsulated plate were similar to those measured on the insulated plate namely, that the tensile strain spectrum shows a peak response at a frequency of about twice the frequency of the peak bending strain, and the tensile strain spectrum was more broadened at 120°F and 160 dB SPL. The probability density distribution of the bending strain response at 120°F also showed one dominant peak prior to snap-through, two peaks during intermittent snap-through, (as the SPL increased) and a plateau between the peaks as the noise level reached 160 dB SPL.

Comparison of Insulated and Uninsulated Plates

Although the insulated plate had a higher edge temperature than that of the uninsulated plate with the same center temperature, the thermal buckling is less. The thermal buckling magnitude is smaller because there are less compressive stresses of the edge of the insulated plate at higher temperatures. Agreement between experimental and theoretical values of thermal buckling displacement is worse for the insulated plate because of the non-ideal clamping condition of the insulated edge. At 160 dB SPL, persistent snap-through motion is obtained for the insulated plate at 150°F, 200°F, and 250°F. However, in both plates the increase of the rms strain response due to snap-through is about 20% of that of the flat plate of ambient temperatures. It should be noted that from reference 5, for an aluminum plate 4" x 8" x 0.040", the rms strain increase due to snap-through motion is 200% for 160 dB SPL.

CONCLUDING REMARKS

Acoustic level of 160 dB can excite intermittent snap-through motion of thermally buckled plates with center deflection twice that of the plate thickness. The snap-through motion had significant effects on the resonant frequencies of the random response summarized as follows:

1. The resonant frequencies decrease with increase of SPL and also increase with the initiation of snap-through.
2. The response peak broadens when persistent snap-through motions are present and the degree of broadening depends on the magnitude of the thermal buckling displacement.
3. The peak responses at low frequencies (1-5 Hz) appear as intermittent snap-through motion starts. The low frequency peaks decrease as persistent snap-through is achieved. The magnitude of the peak response depends on the magnitude of the thermal buckling displacement.

For persistent snap-through motion of the uninsulated plate at 120°F and 160 dB SPL, the rms strain response was only 20% higher than that of the flat plate. This small increase resulted because the 160 dB SPL can only excite persistent snap-through motion of the thermally buckled plate at 120°F, for which the center deflection is only 62% of the thickness. For higher buckling deflection of the plate at higher temperatures, the stiffening effect of the curvature becomes more important and the overall acoustic response decreases considerably.

From the above results, it can be concluded that the amount of increase or decrease of acoustic response of a heated plate compared to that of a plate at ambient temperature depends on the magnitude of thermal buckling and the ability of the acoustic loads to excite snap-through motion.

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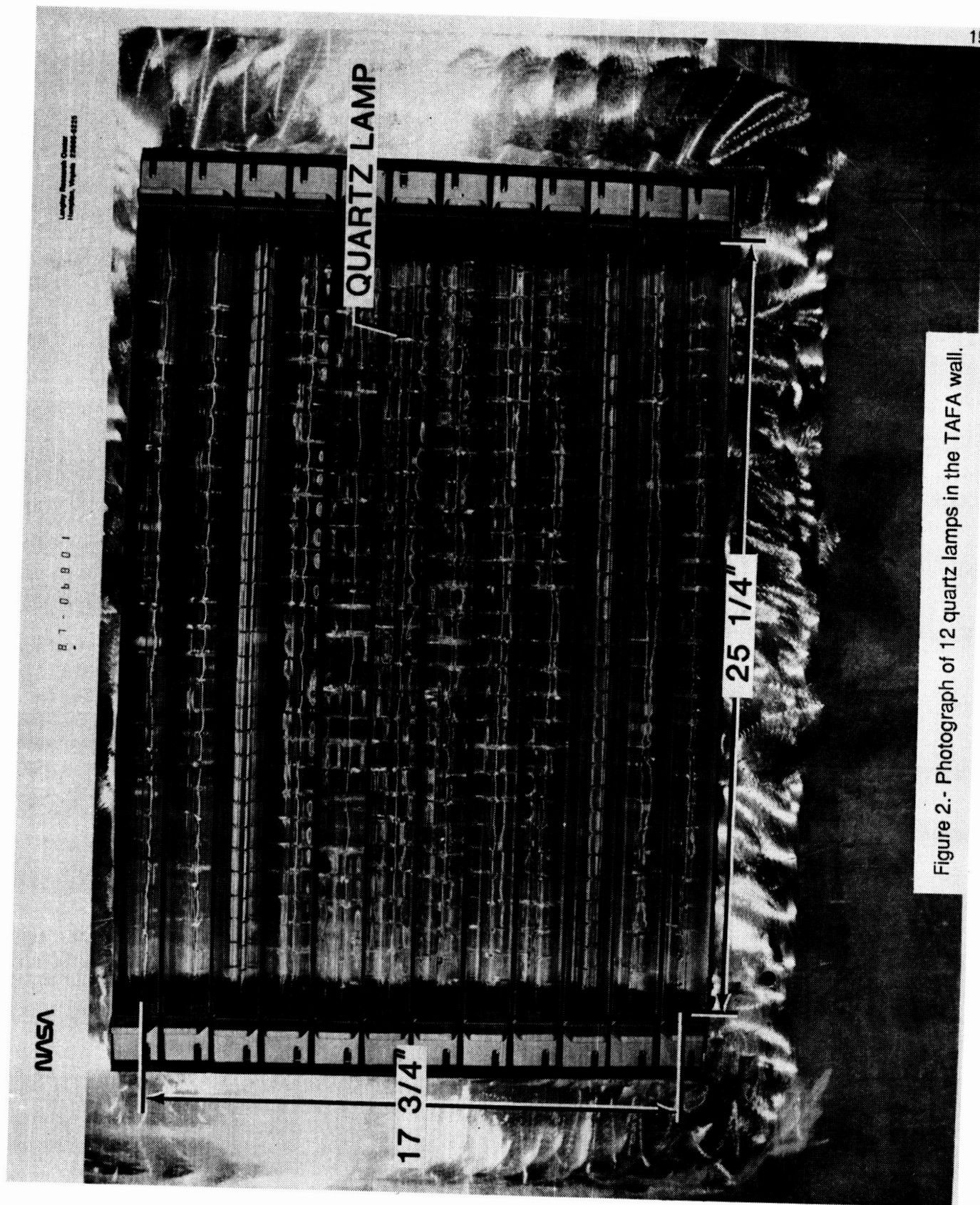
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Figure 1.- Photograph of the Thermal Acoustic Fatigue Apparatus.

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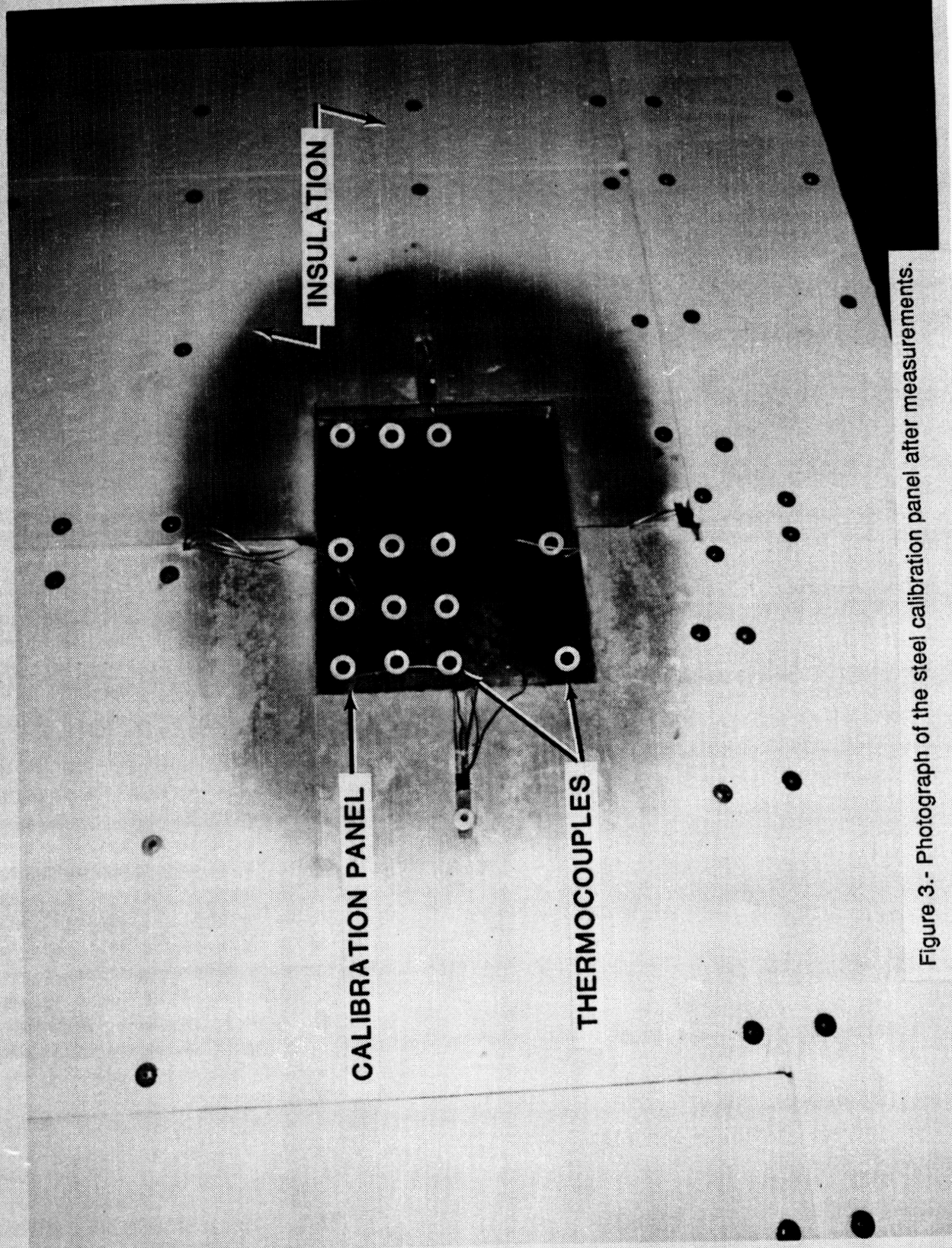


Figure 3.- Photograph of the steel calibration panel after measurements.

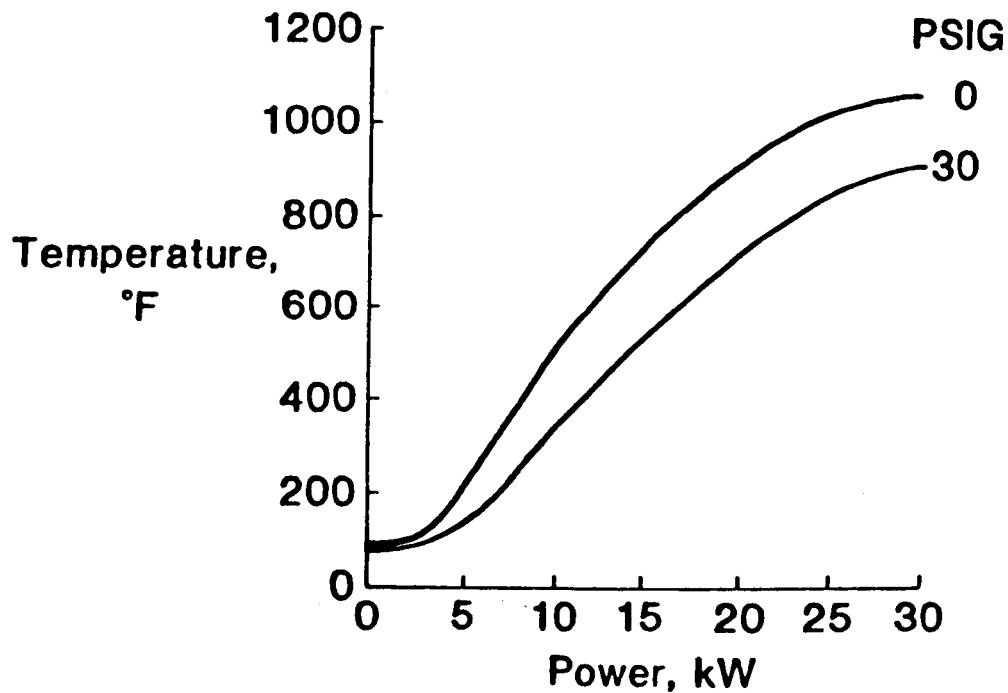
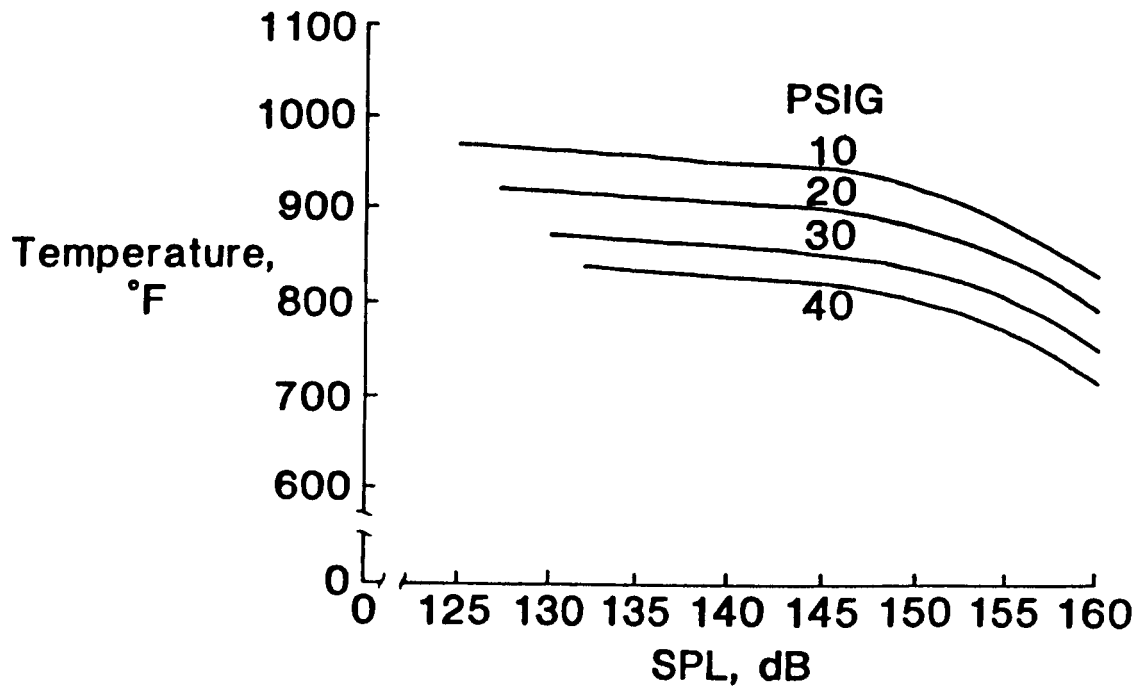


Figure 4.- Effects of airflow, sound pressure level and electrical power on temperature of calibration panel.

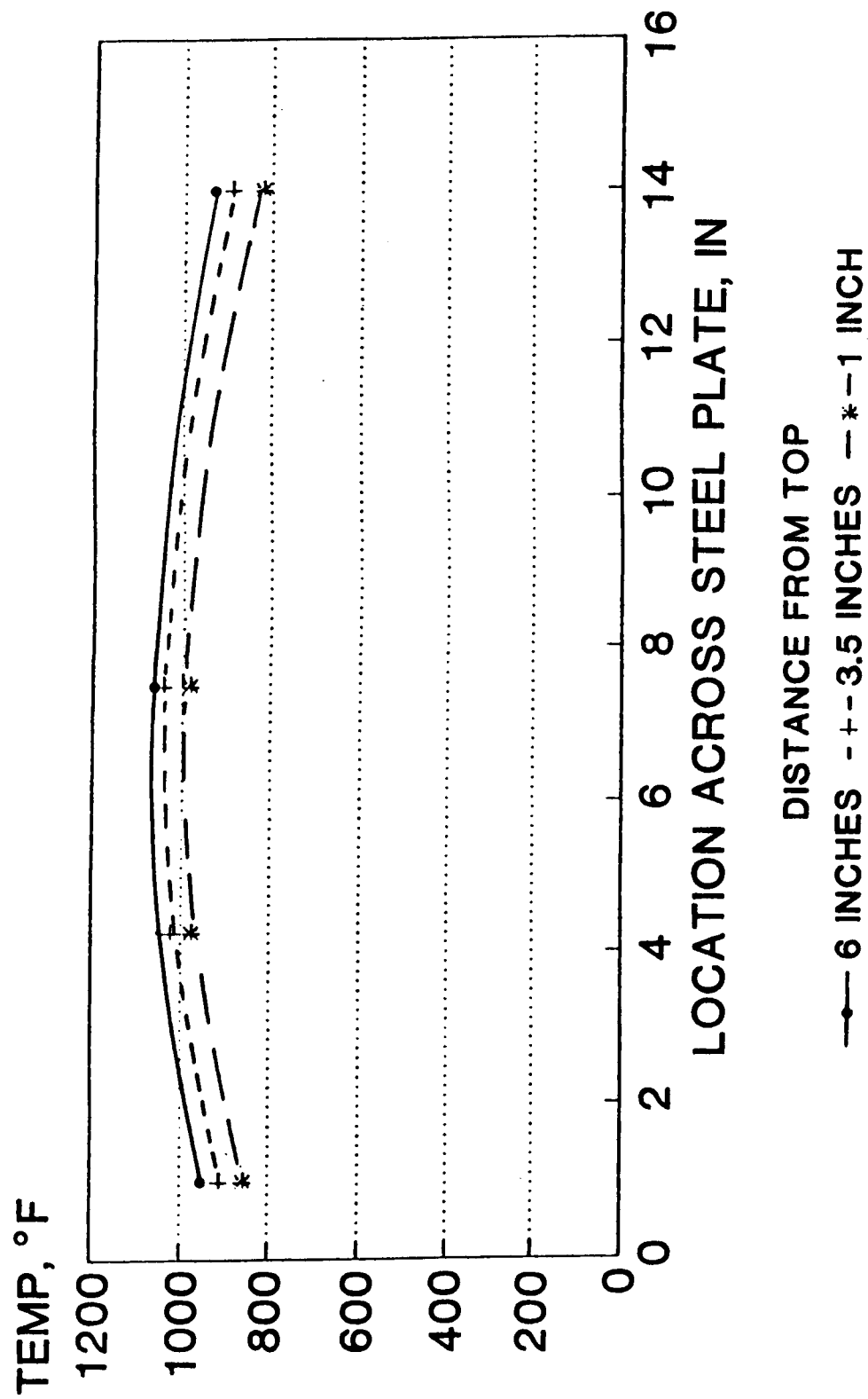
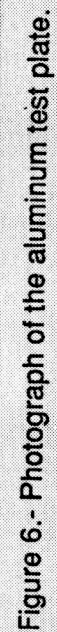


Figure 5.- Temperature distribution of the top half of the steel plate. No air flow.



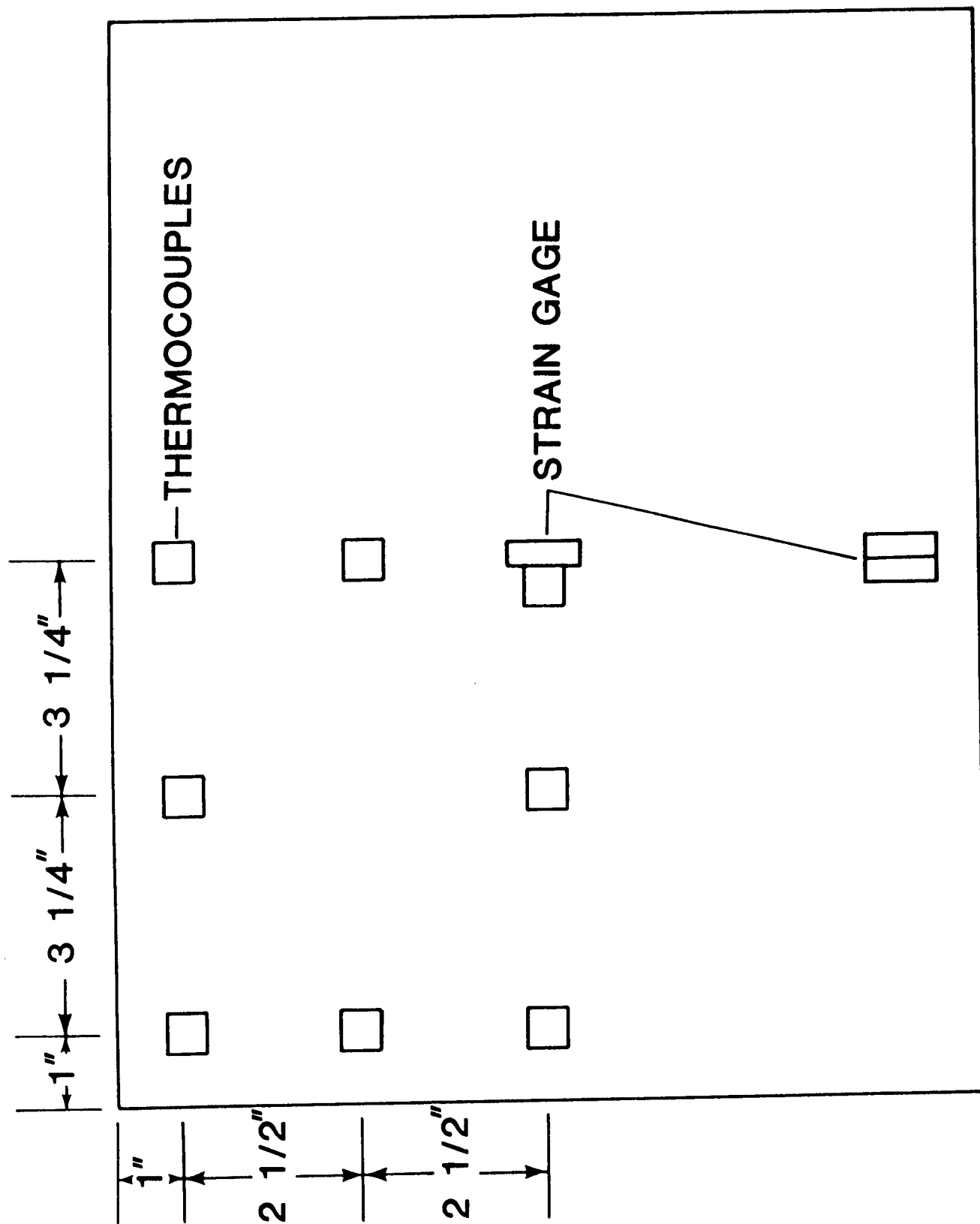


Figure 7.- Location of strain gages and thermocouples on the aluminum plate.

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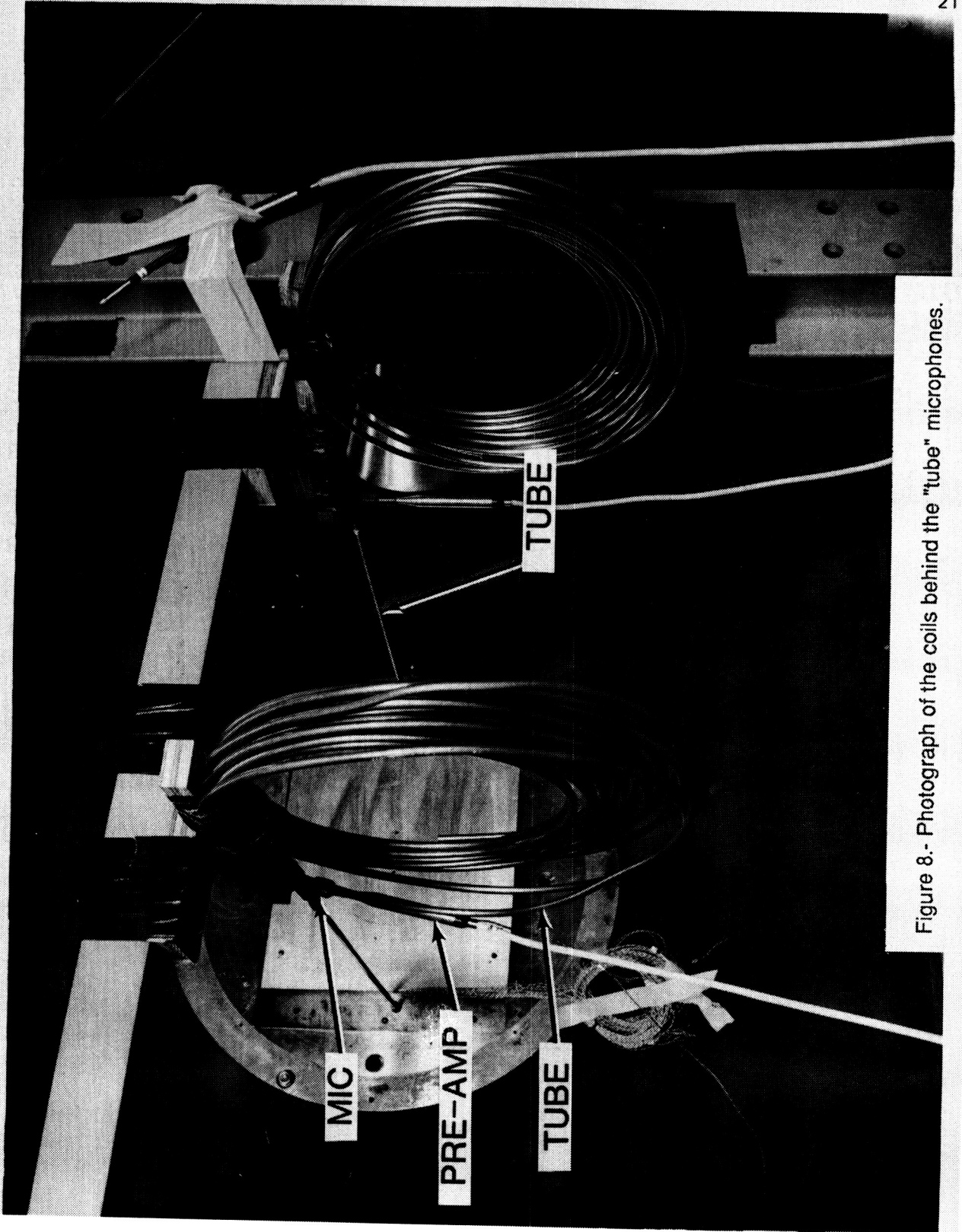


Figure 8.- Photograph of the coils behind the "tube" microphones.

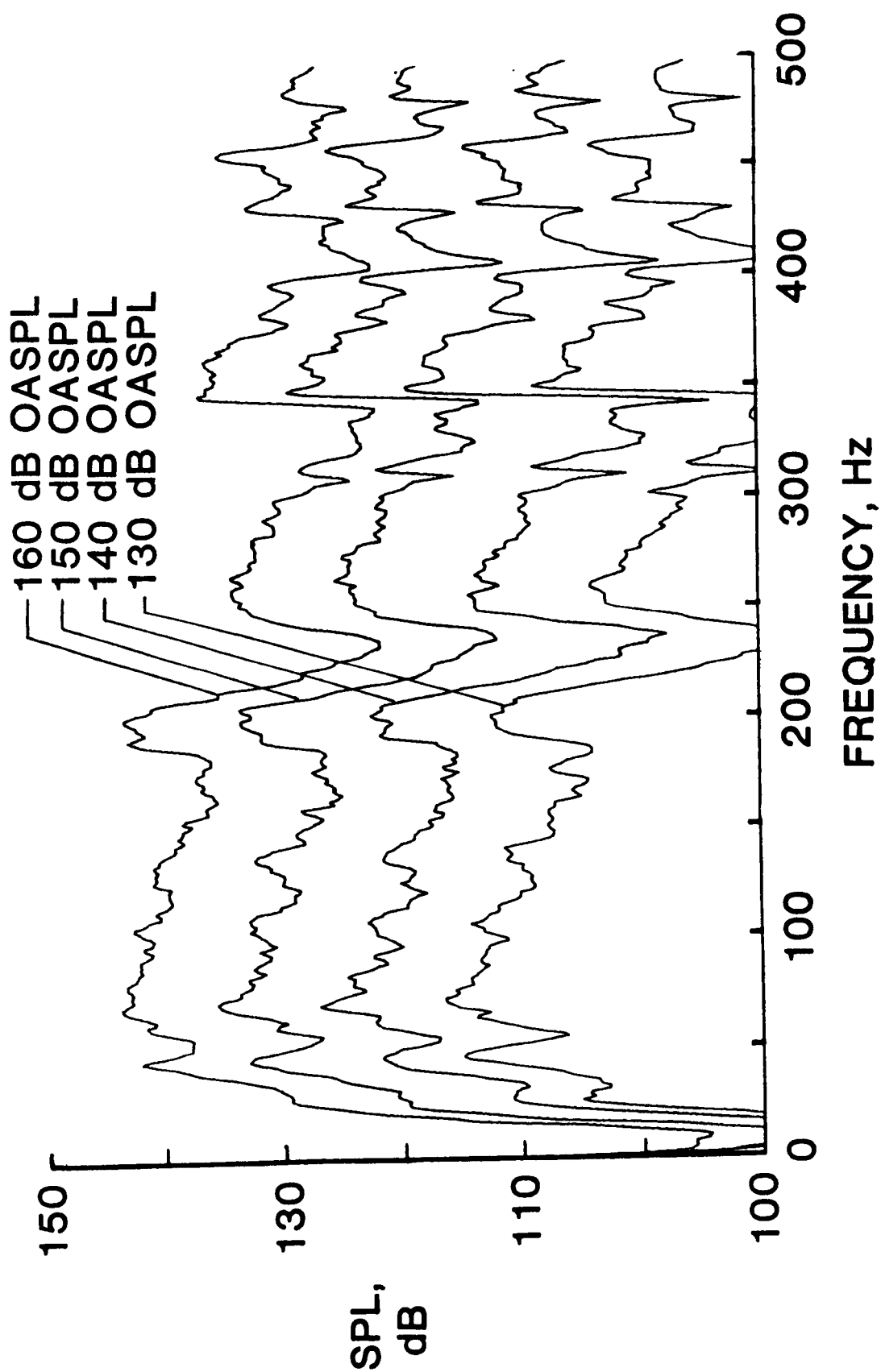


Figure 9.- Acoustic spectra to which the aluminum plates were subjected.

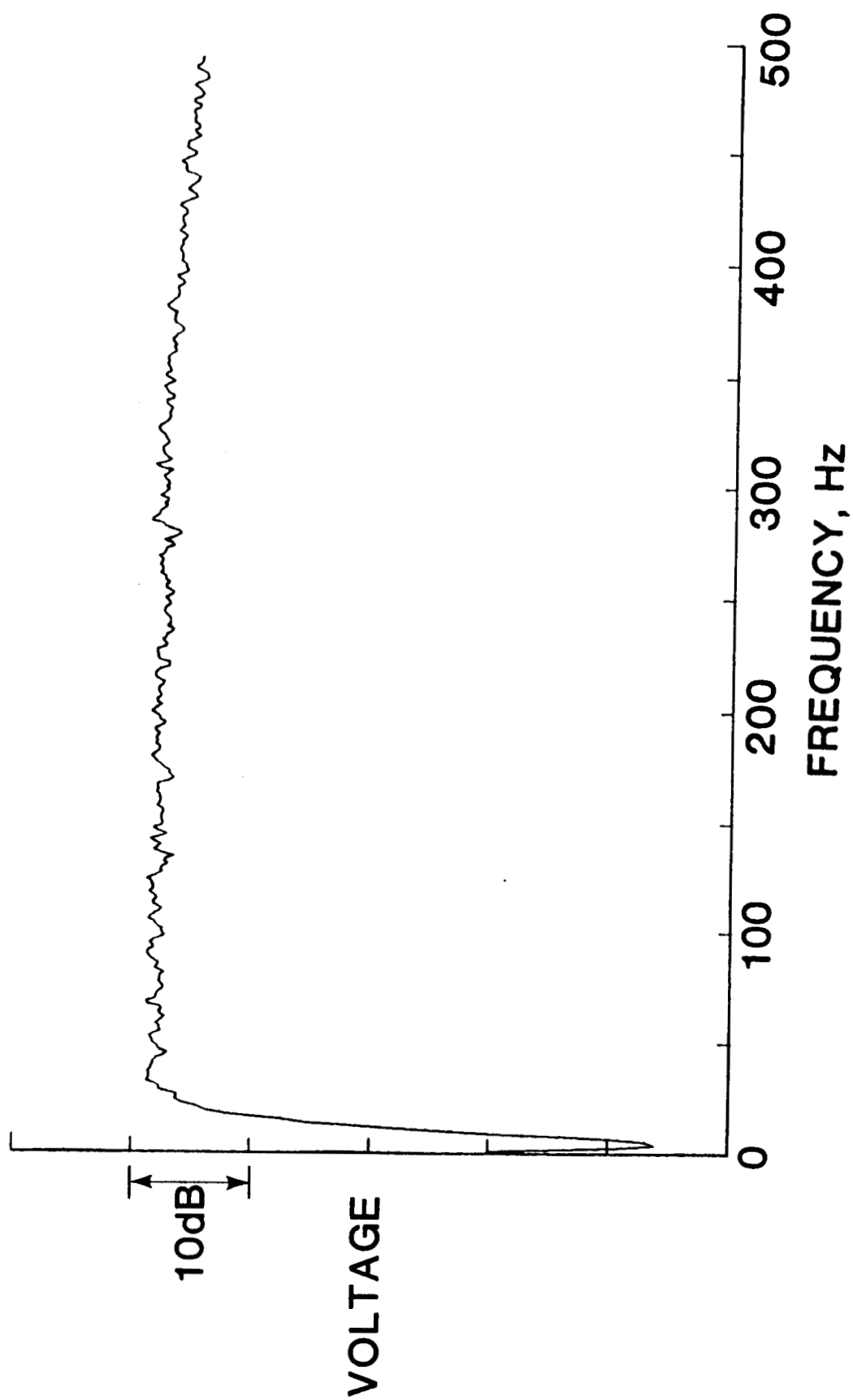


Figure 10.- Input spectrum to drive the modulators.

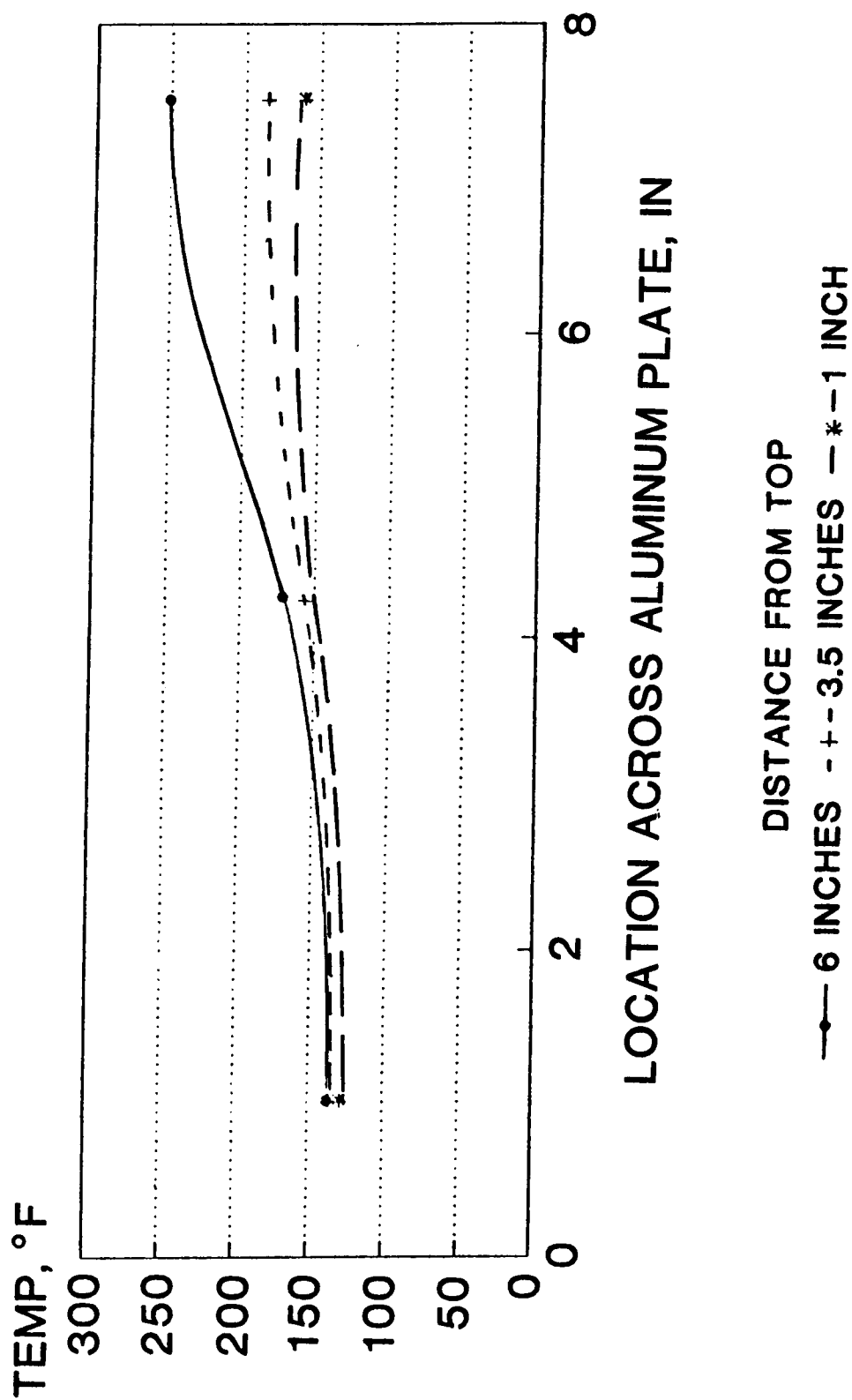


Figure 11.- Temperature distribution over one quadrant of the aluminum plate with no insulation.

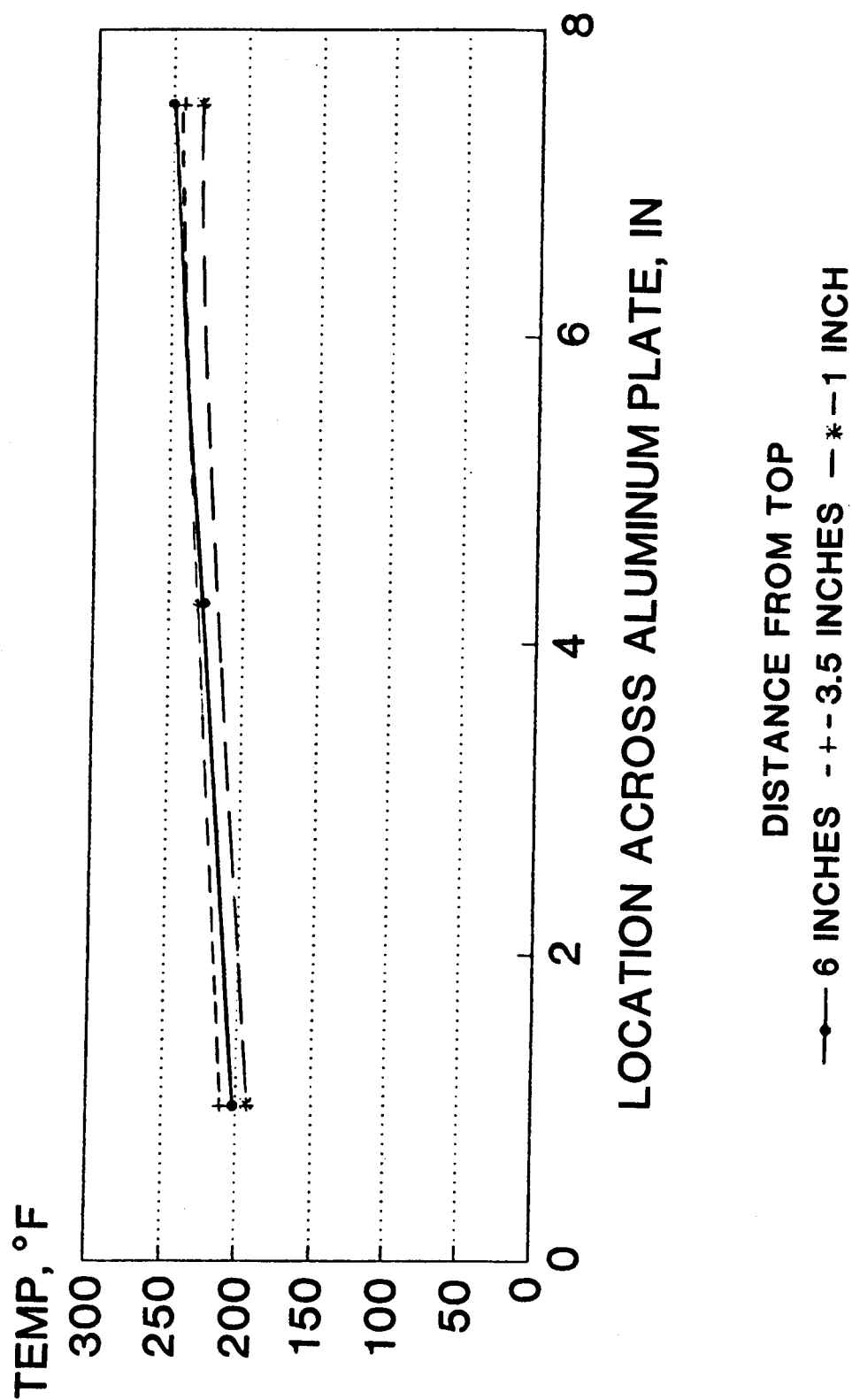


Figure 12.- Temperature distribution over one quadrant of the aluminum plate with fiberglass insulation.

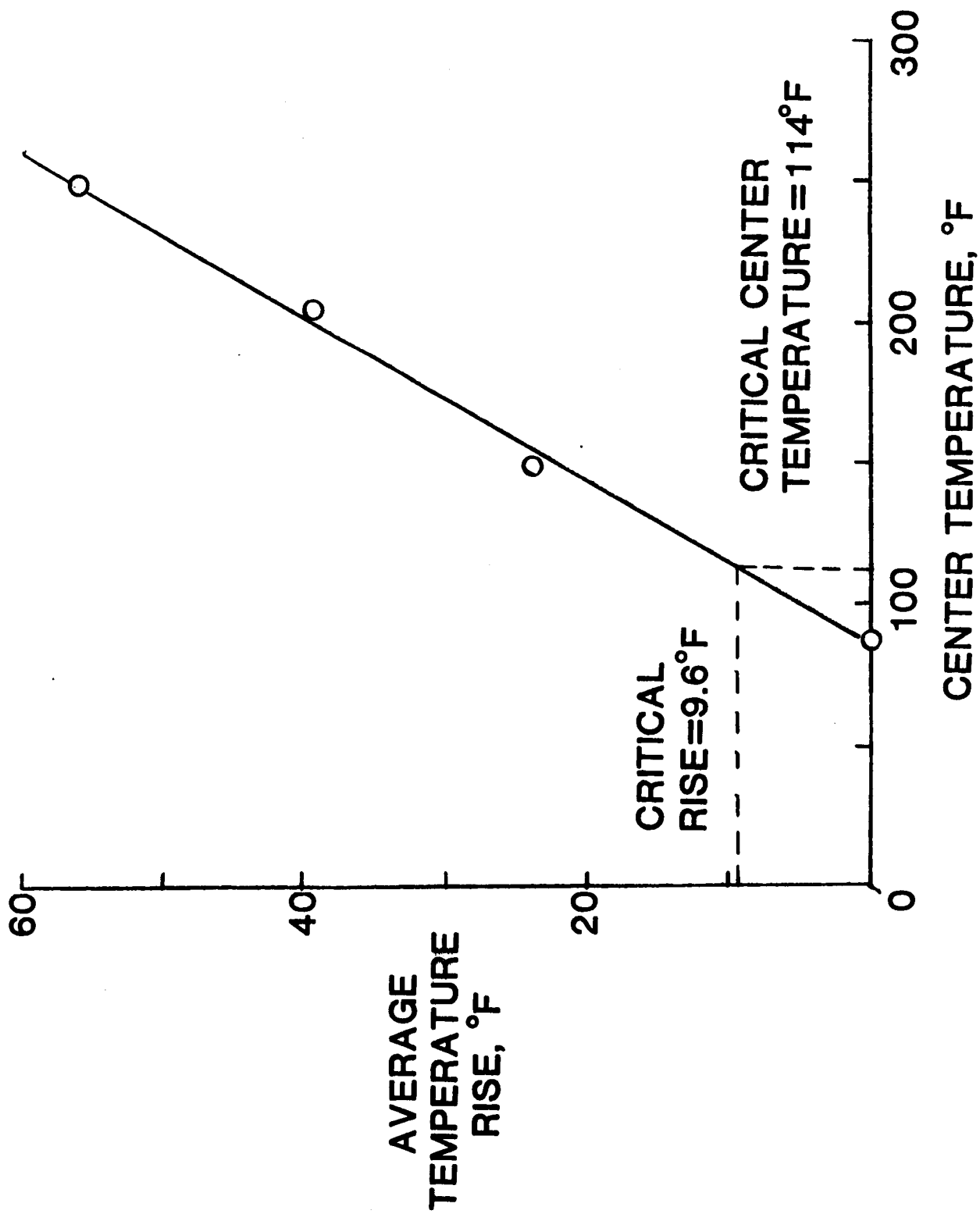


Figure 13.- Average temperature rise above the edge temperature as a function of plate center temperature for the uninsulated plate.

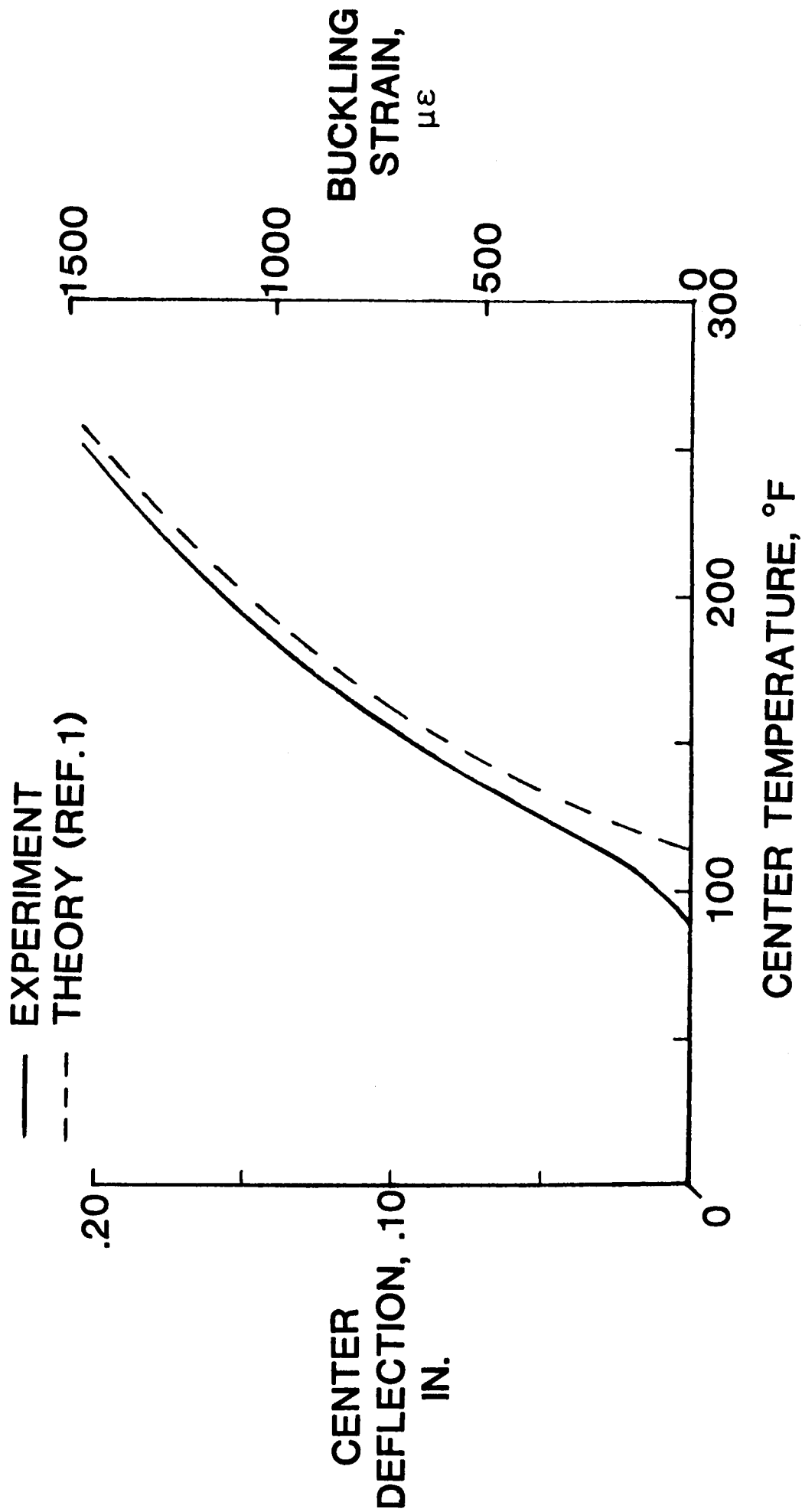


Figure 14.- Center deflection due to buckling as a function of plate center temperature.

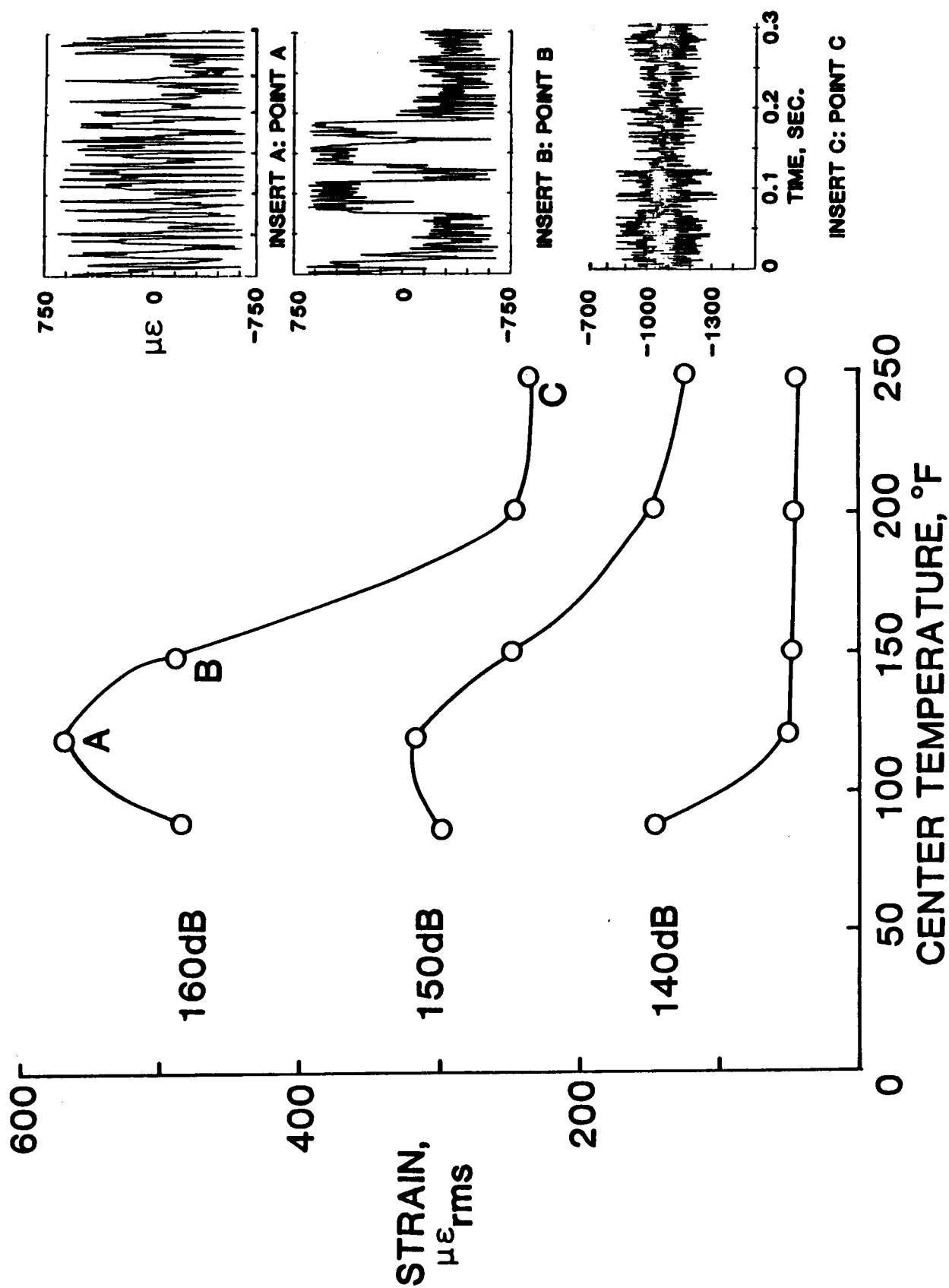


Figure 15.- Acoustic response at various SPL's as a function of plate center temperature with inserts of various motions.

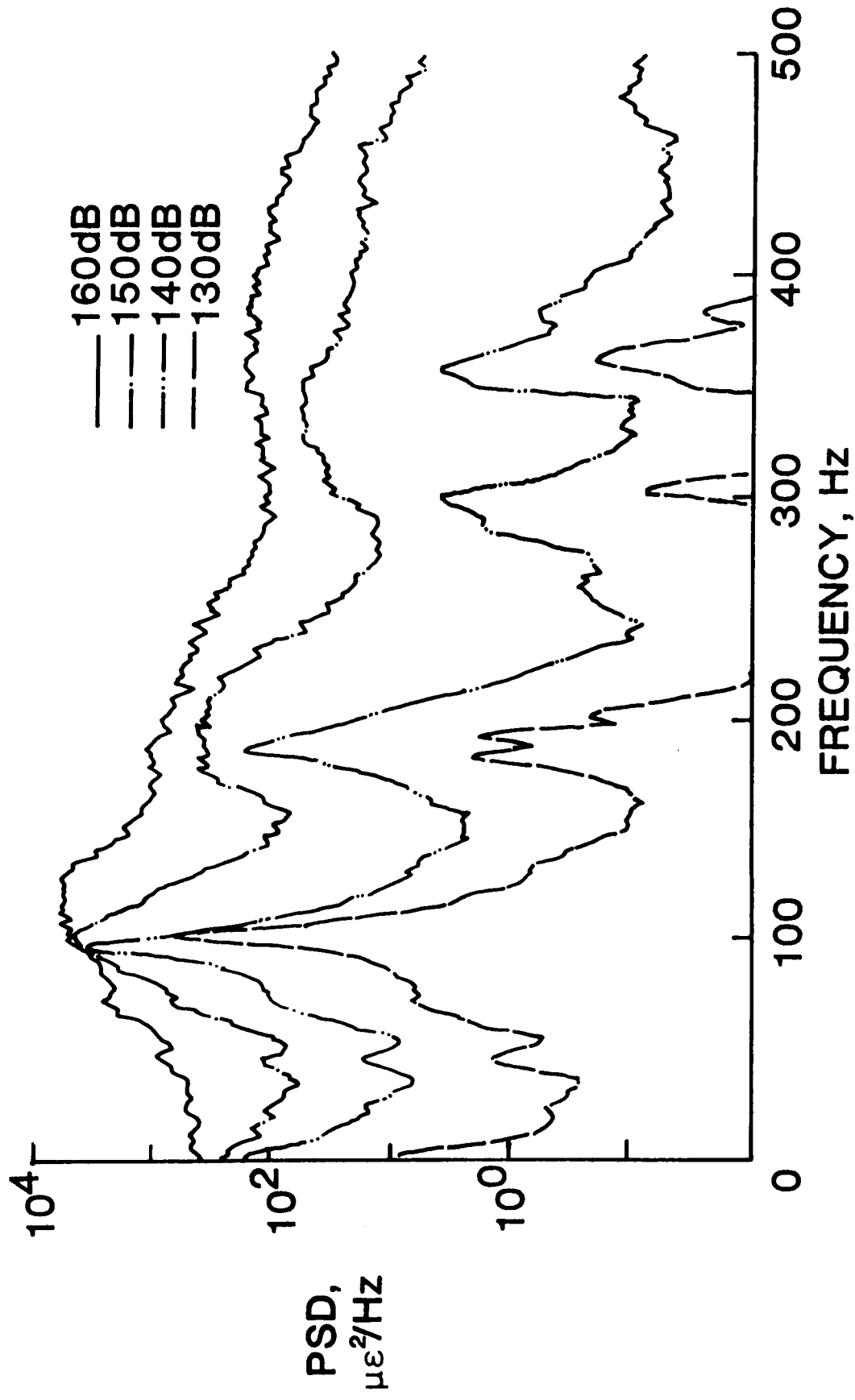


Figure 16.- Power spectral density (PSD) of the bending strain at various SPL's at ambient temperature (87°F).

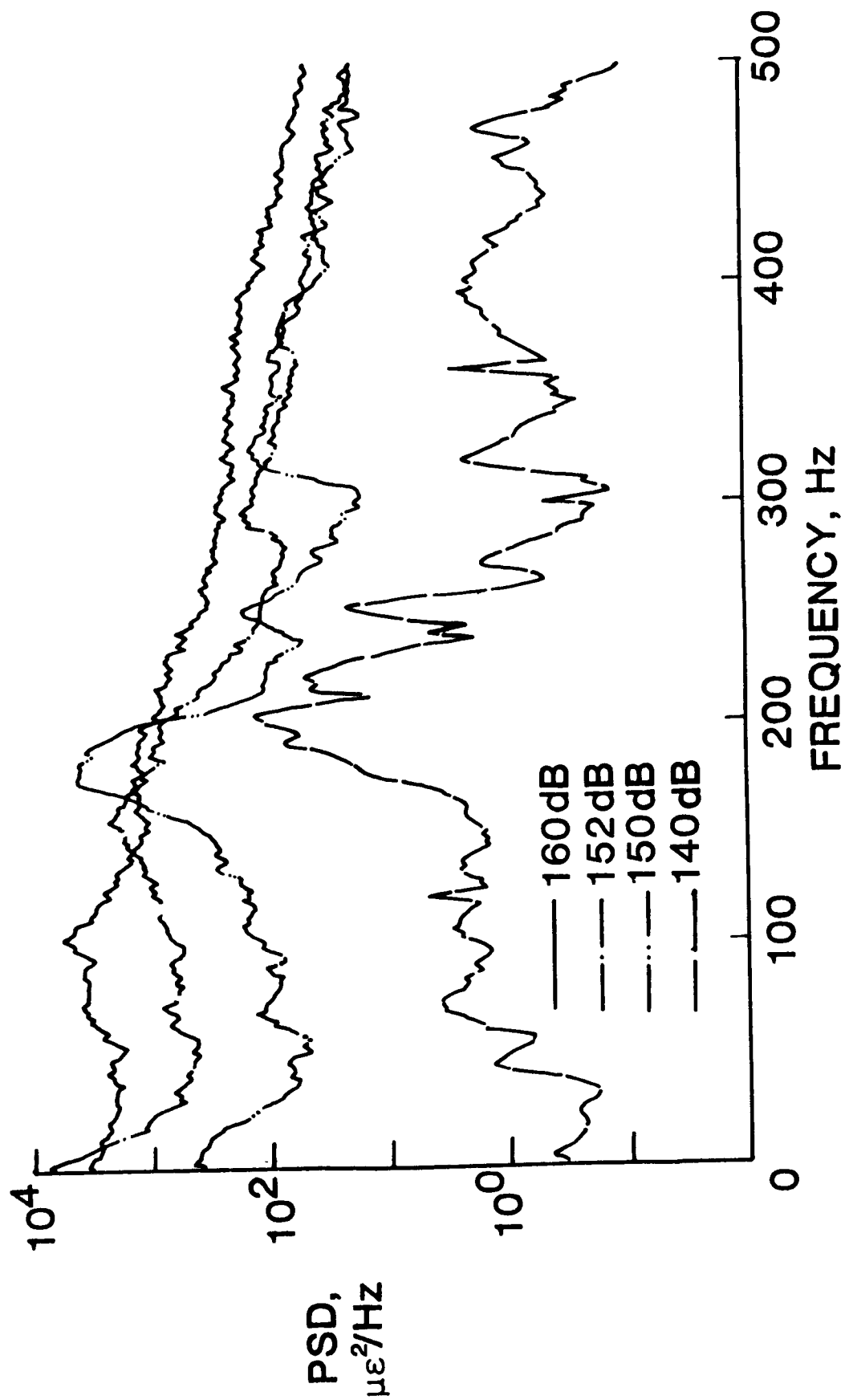


Figure 17.- Power spectral density (PSD) of the bending strain at various SPL's at 120°F.

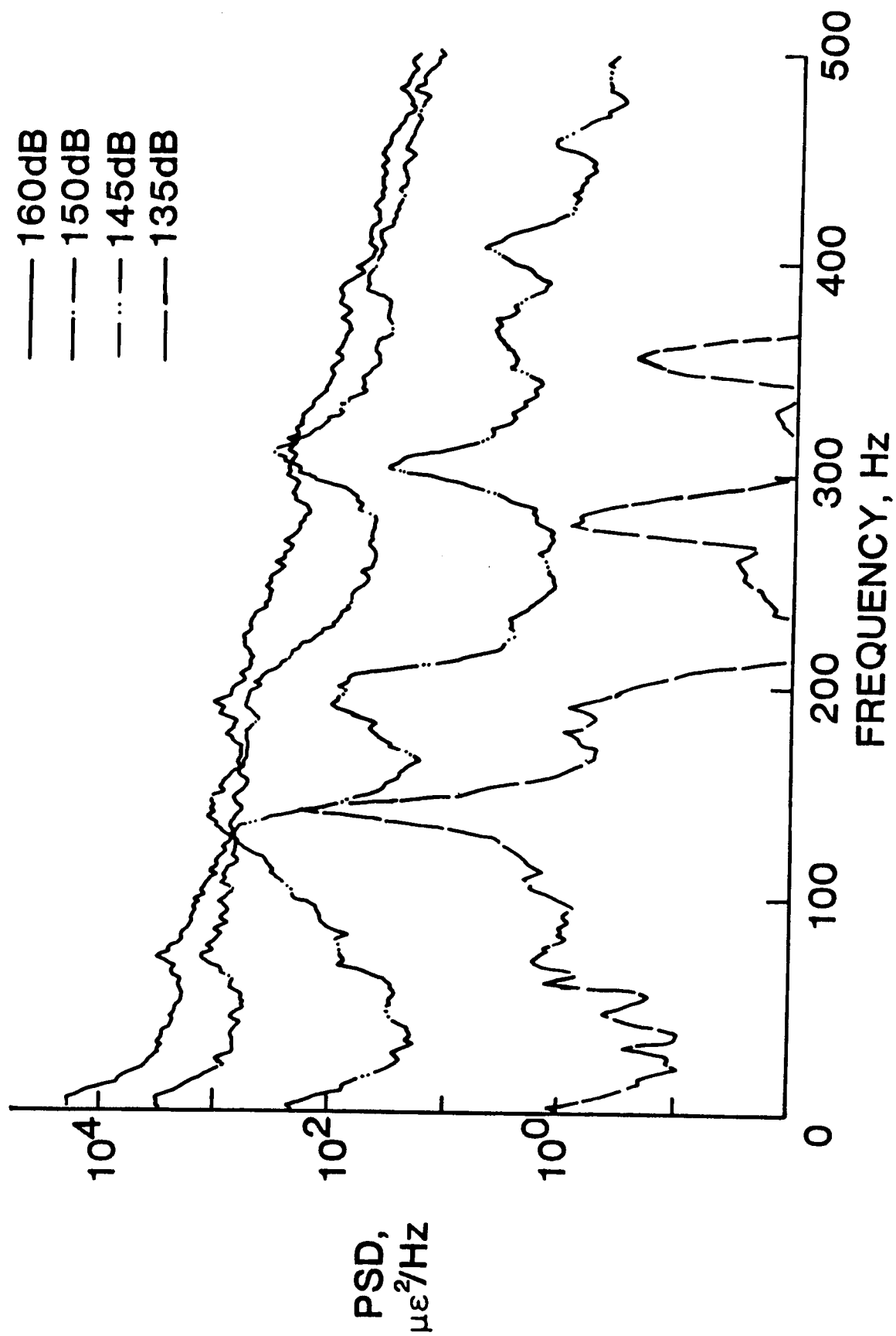


Figure 18.- Power spectral density (PSD) of the bending strain at various SPL's at 150°F.

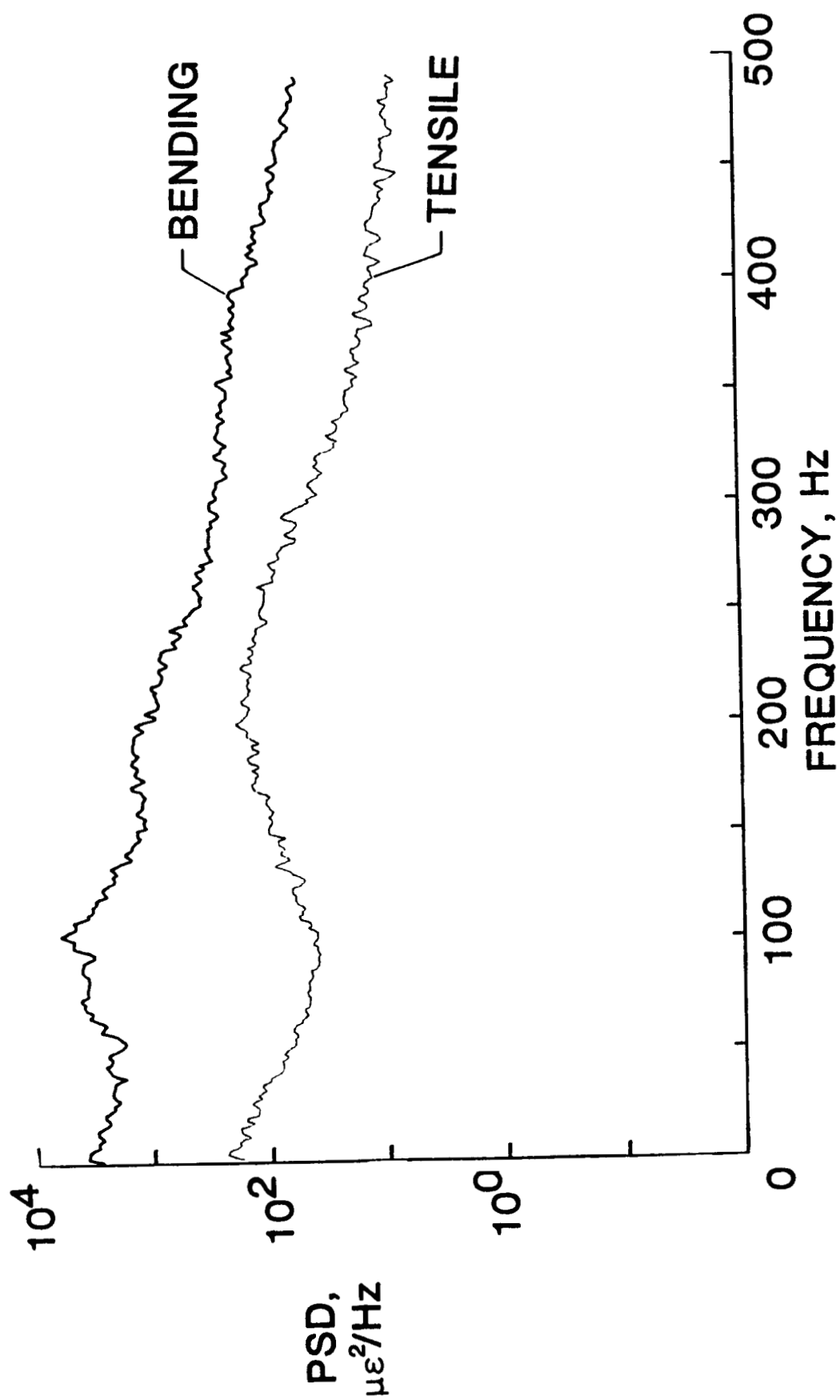


Figure 19.- Comparison of bending and tensile PSD's at 120°F for 160 dB SPL.

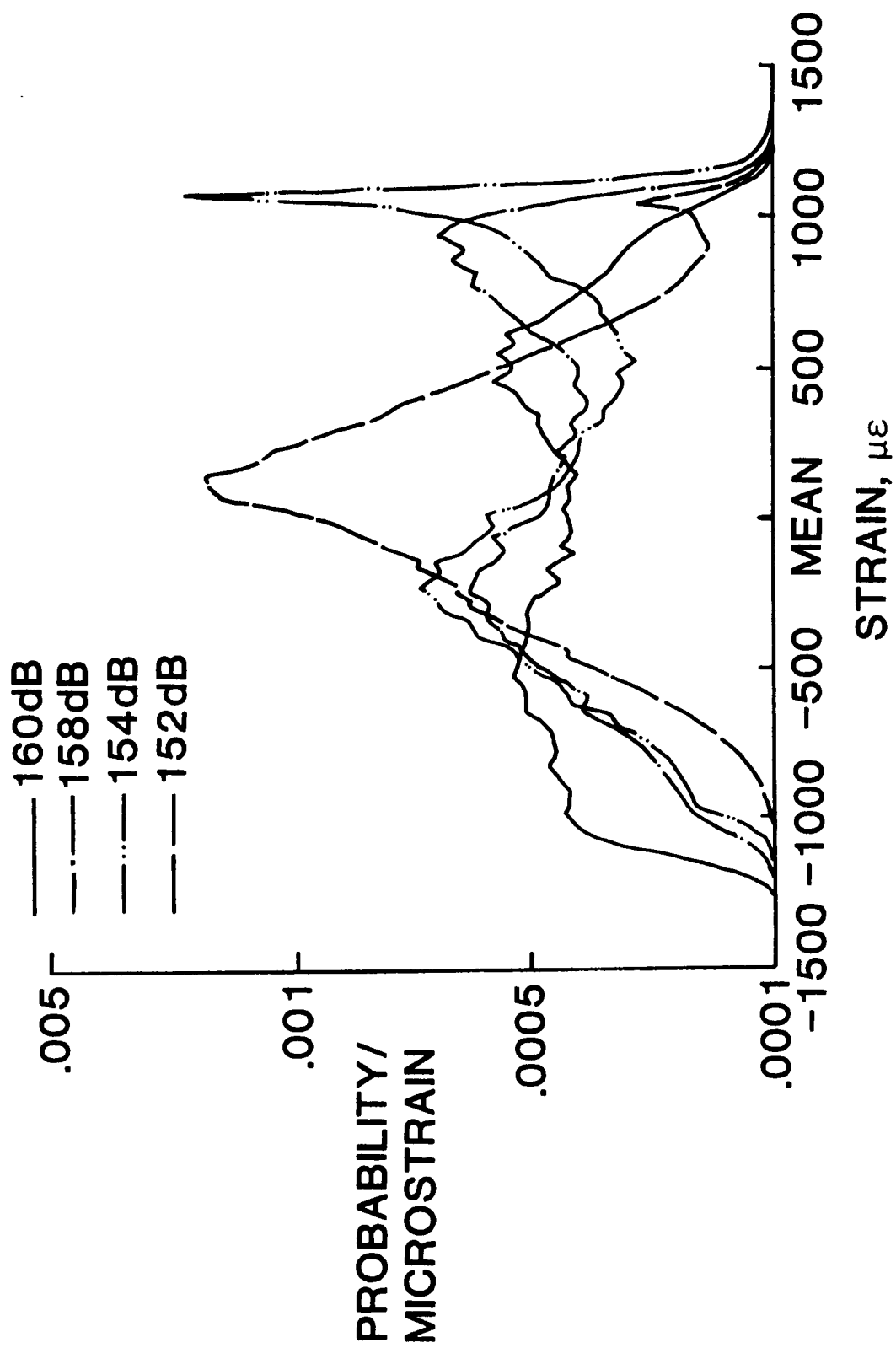


Figure 20.- Probability density distribution of bending strain responses at 120°F for various SPL's.

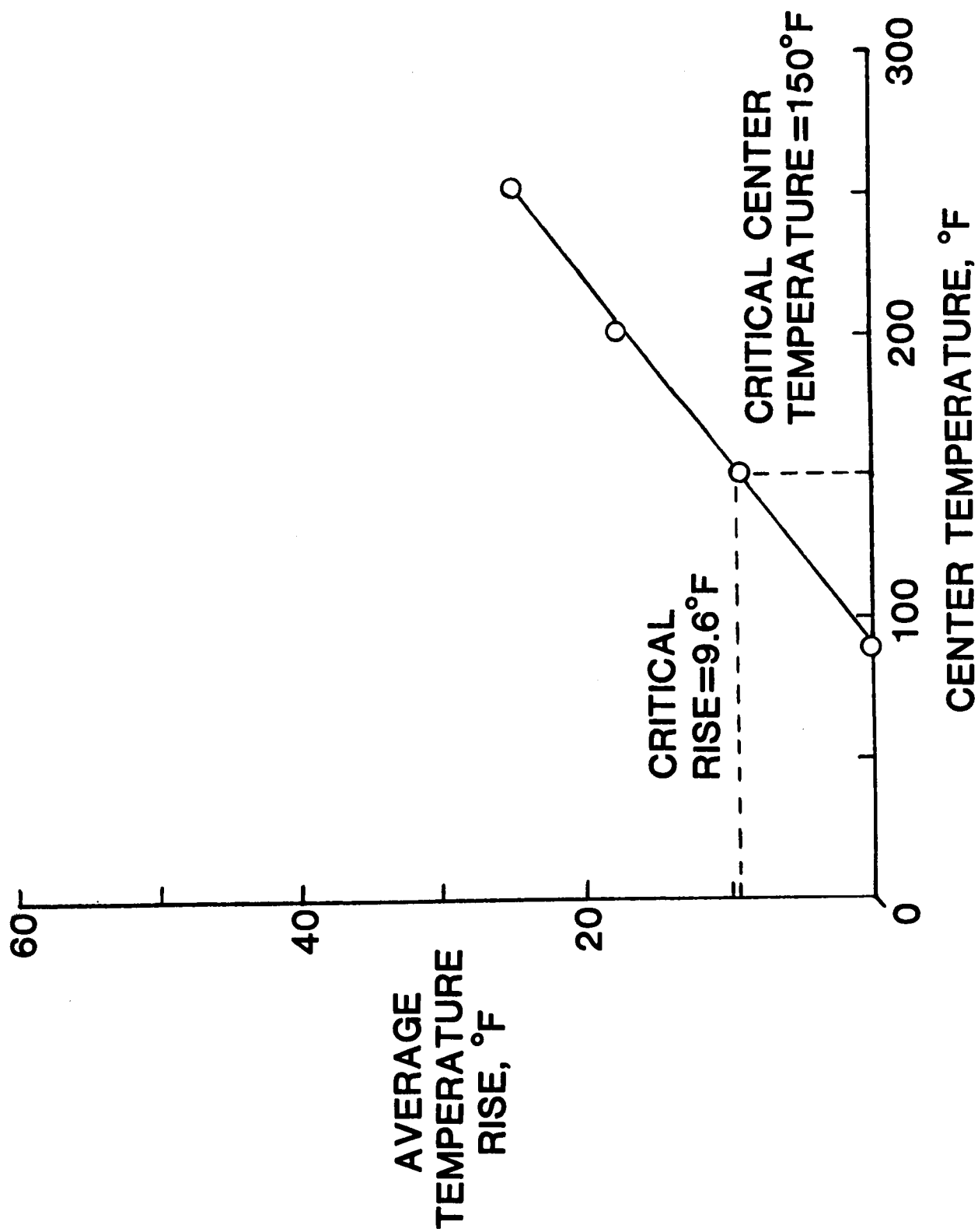


Figure 21.- Average temperature rise above the edge temperature as a function of plate center temperature for the insulated plate.

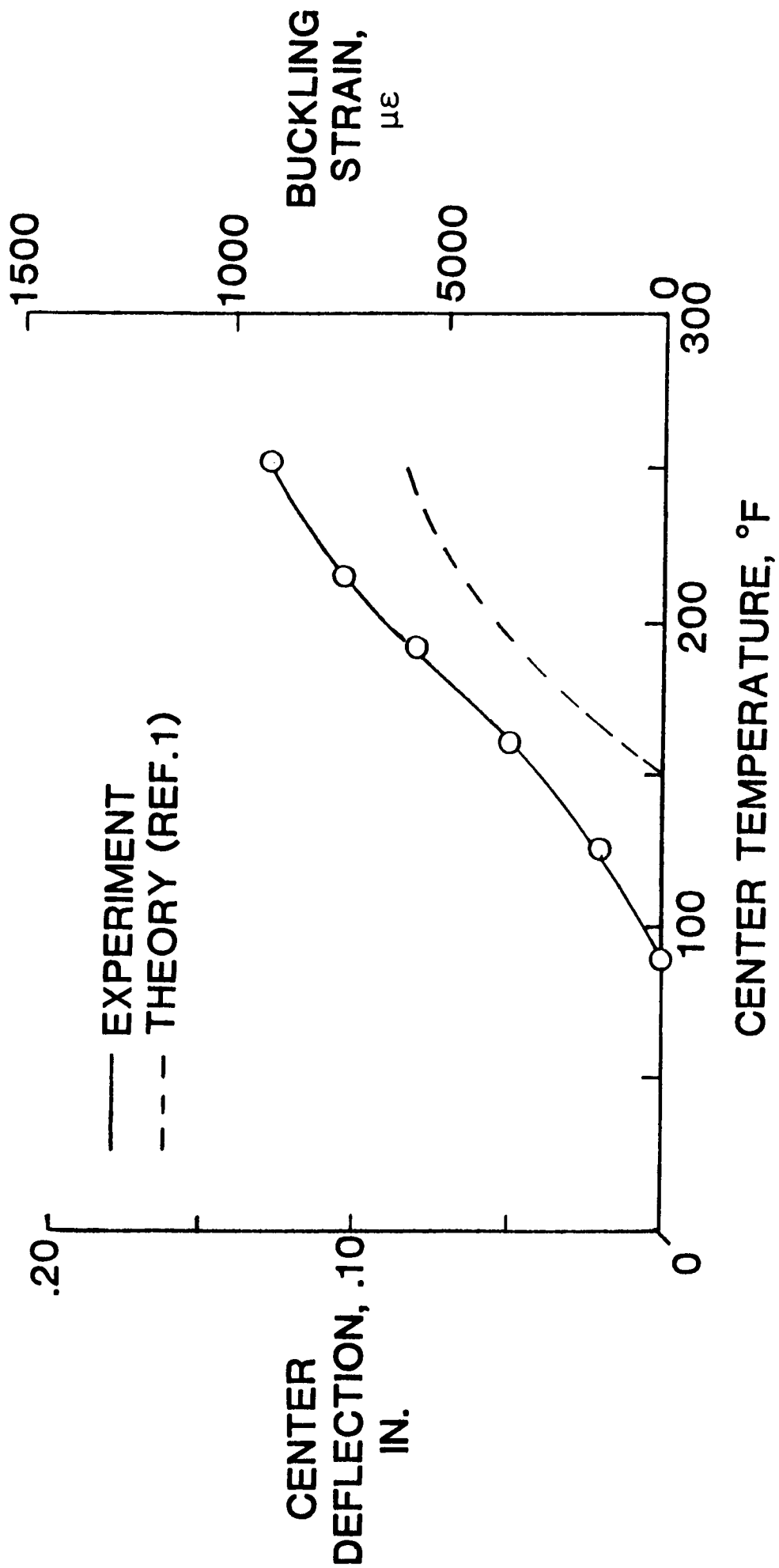


Figure 22.- Center deflection due to buckling as a function of plate center temperature.

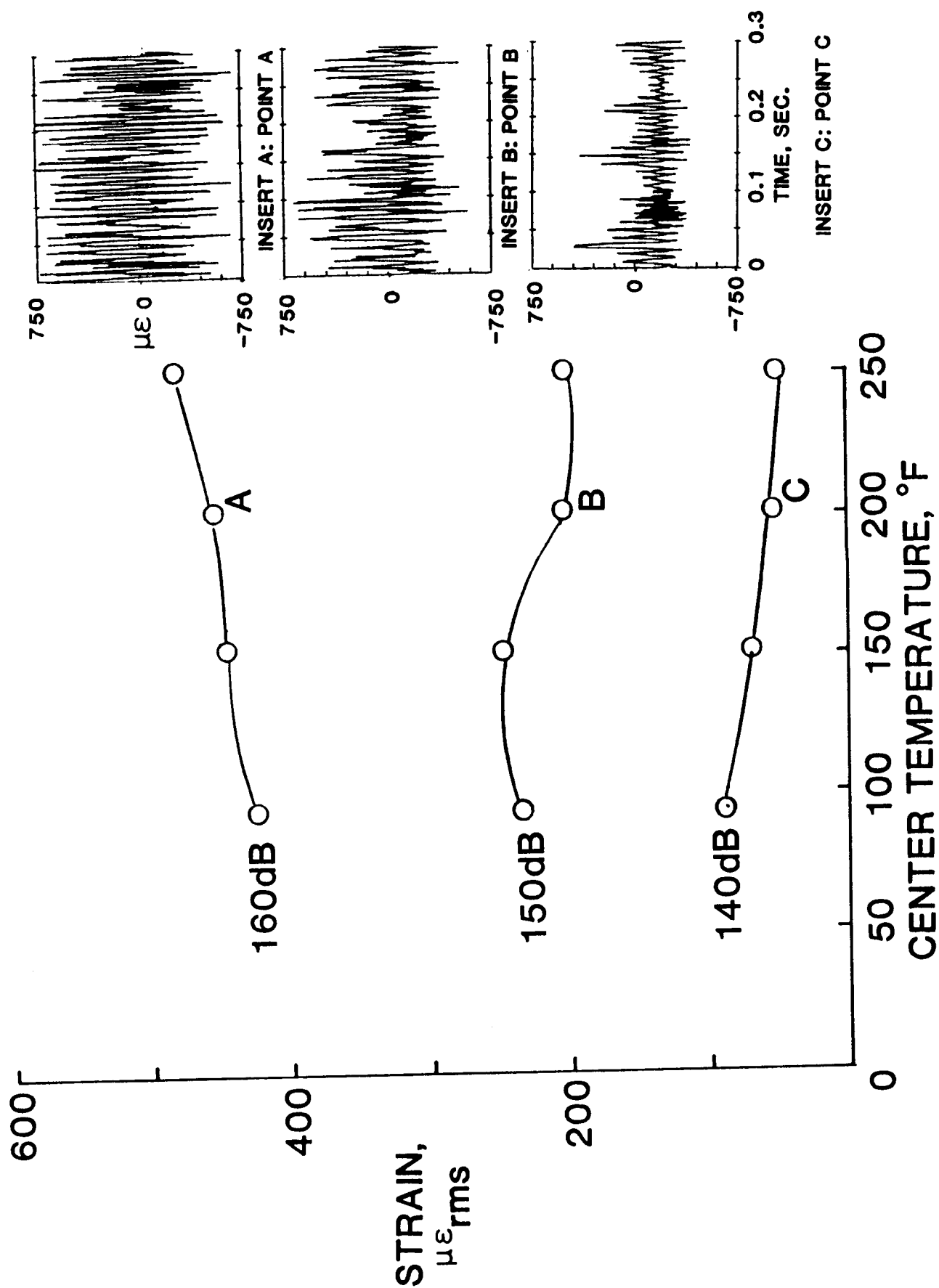


Figure 23.- Acoustic response at various SPL's as a function of plate center temperature with inserts of various motions.

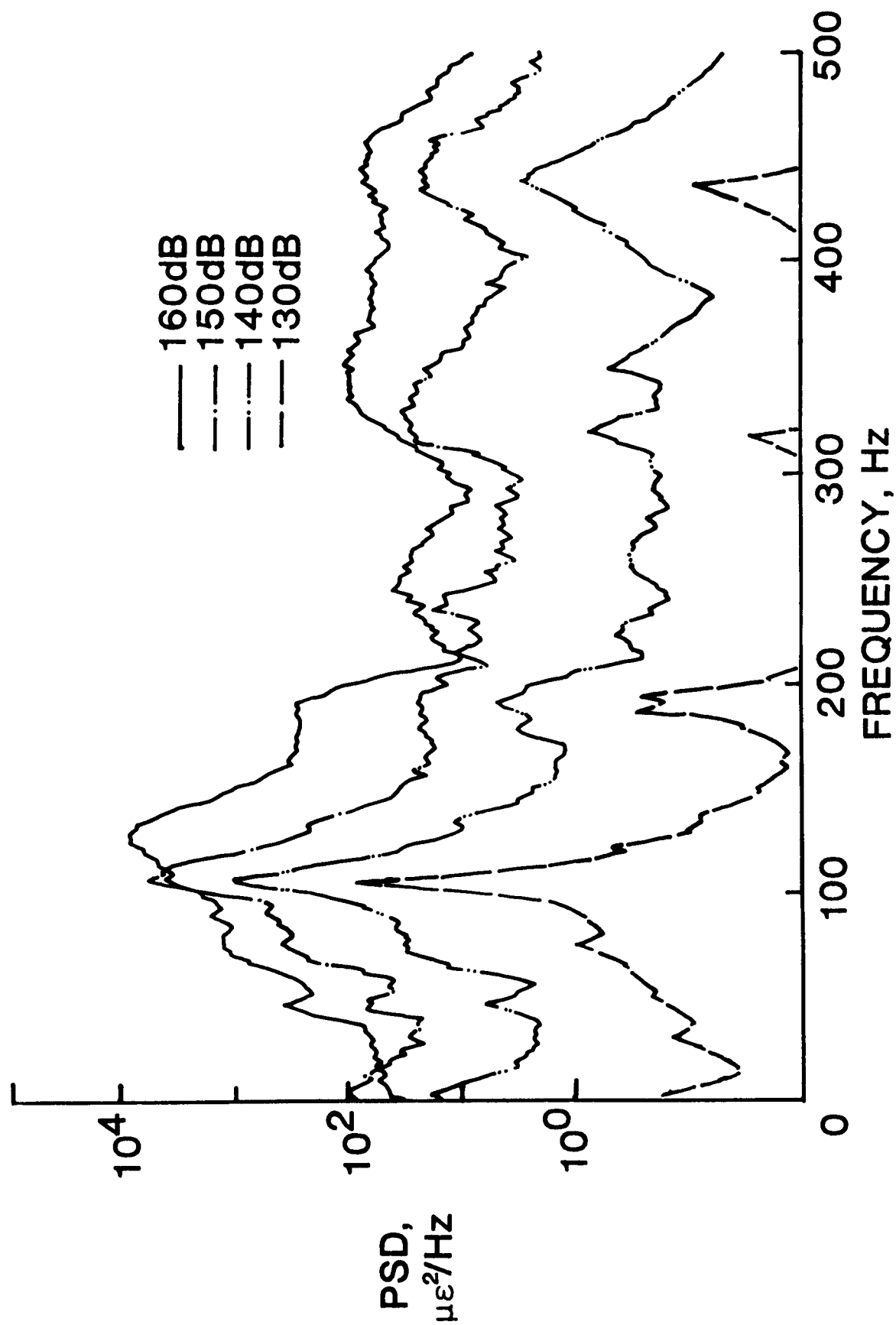


Figure 24.- PSD's of the bending strains for various SPL's at ambient temperature (89°F).

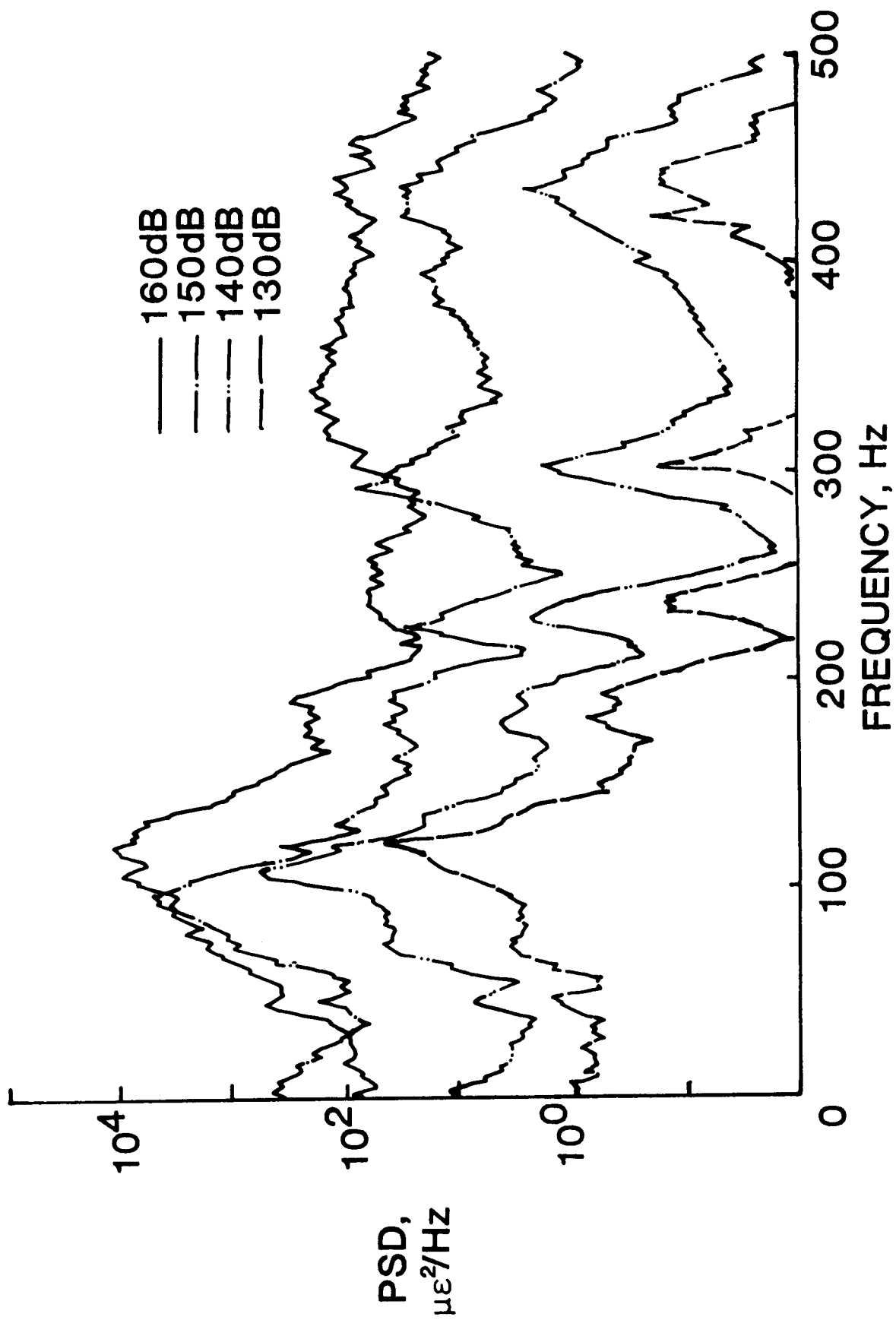


Figure 25.- PSD's of the bending strains for various SPL's at 150°F.

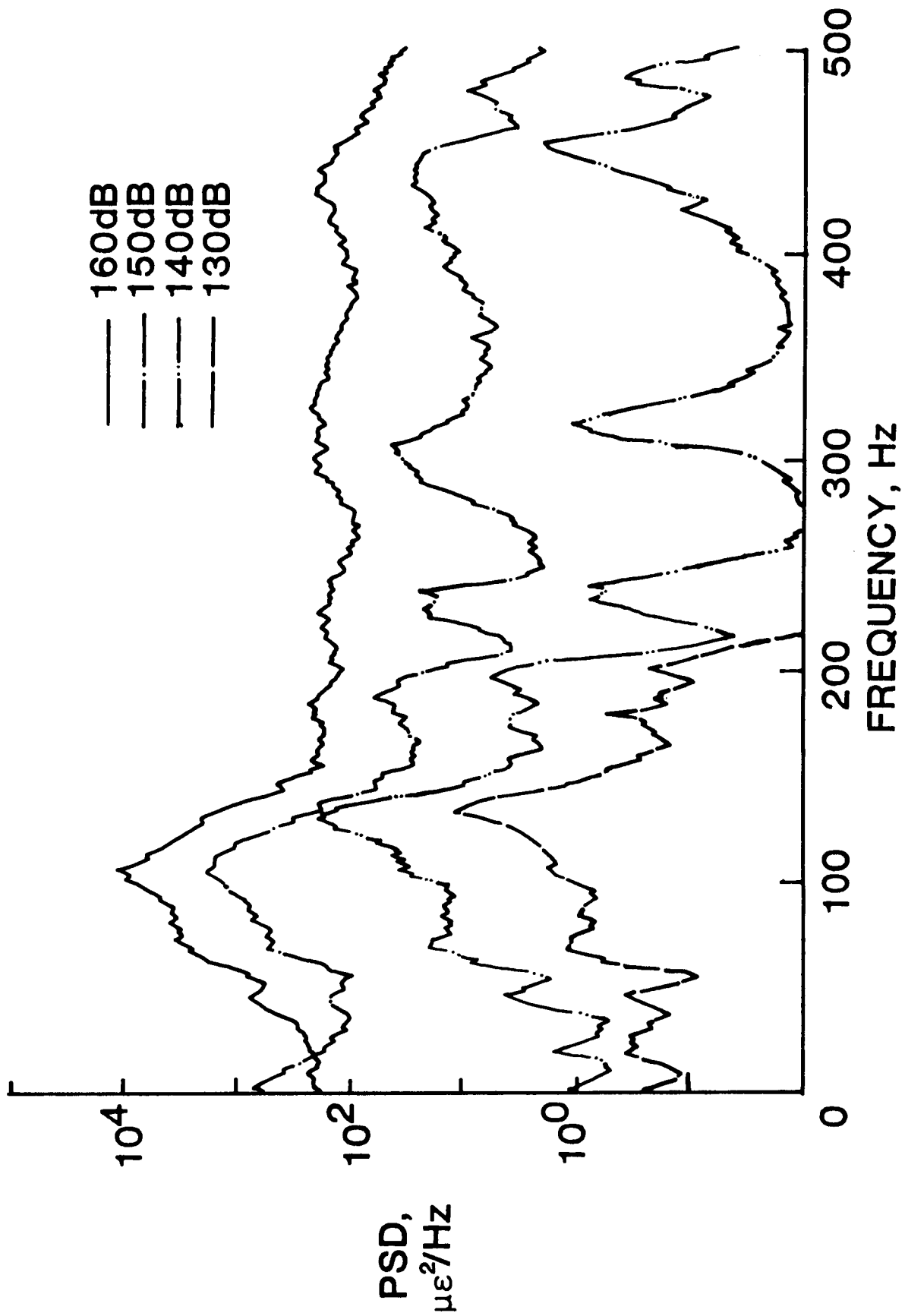


Figure 26.- PSD's of the bending strains for various SPL's at 200°F.

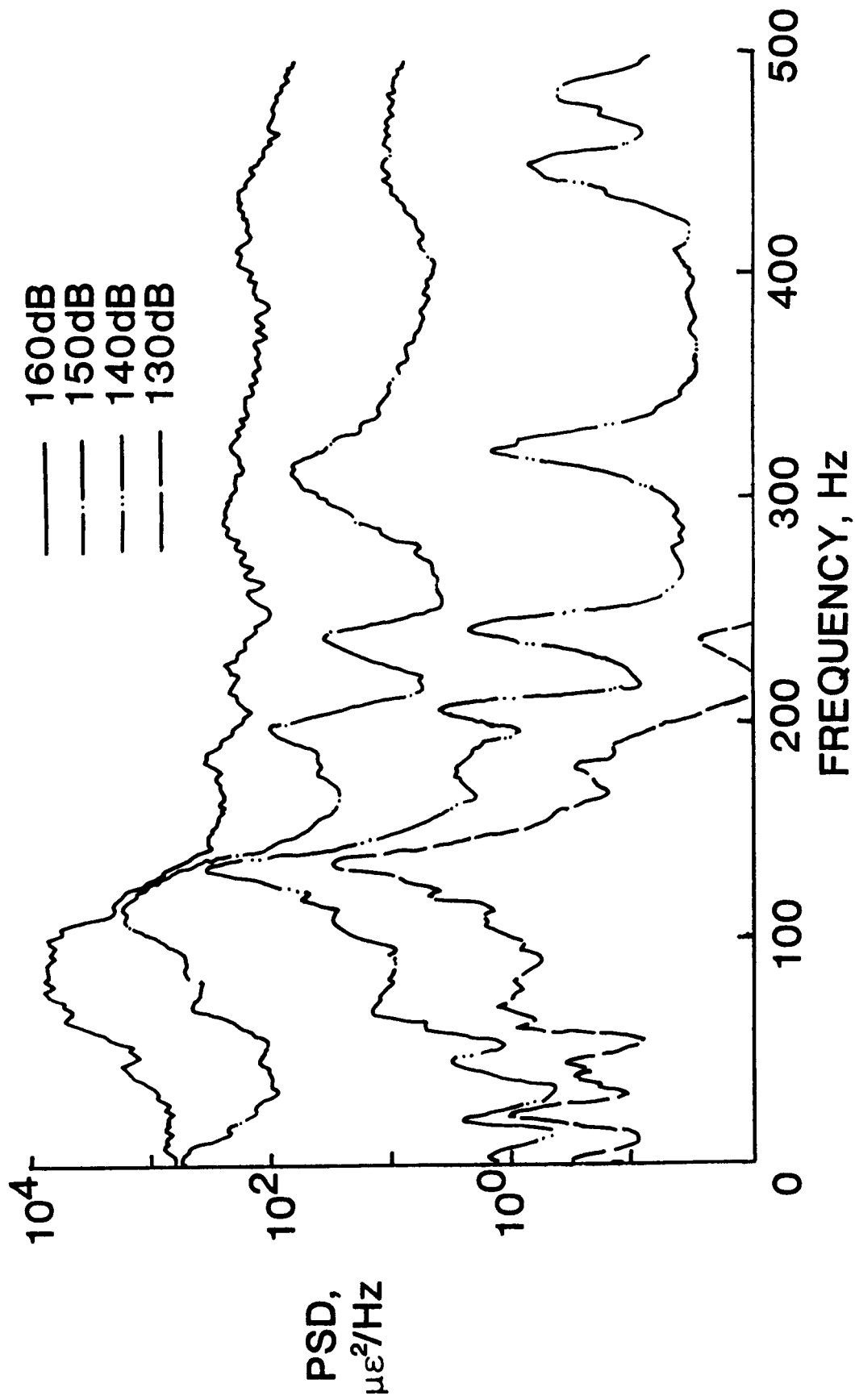


Figure 27.- PSD's of the bending strains for various SPL's at 250°F.



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